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SACRAMENTO PLANT

INVESTIGATION OF SYSTEM COMPONENTS FOR HIGH CHAMBER PRESSURE ENGINES

Final Report

Contract NAS 8-5329

Part 2: Appendixes

Report 5329-F

May 1964

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AEROJET GENERAL TIRE GENERAL

AEROJET-GENERAL CORPORATION

SACRAMENTO, CALIFORNIA

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INVESTIGATION OF SYSTEM COMPONENTS FOR HIGH CHAMBER PRESSURE ENGINES

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Prepared under

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Under Technical Direction of

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Huntsville, Alabama

AEROJET-GENERAL CORPORATION
A SUBSIDIARY OF THE GENERAL TIRE & RUBBER COMPANY

APPENDIX A

GOVERNMENT CONTRACTS APPLICABLE TO SEAL TECHNOLOGY

GOVERNMENT CONTRACTS APPLICABLE TO SEAL TECHNOLOGY

NAS 7-102 "Static and Dynamic Seals for Liquid Rocket Engines," General Electric Advanced Technology Laboratories, Schenectady, New York.

NAS 7-107 "Advanced Valve Technology for Spacecraft Engines," TRW Space Technology Laboratories, Redondo Beach, California.

NAS 8-4012 "Design Criteria for Zero-Leakage Connectors for Launch Vehicles," General Electric Advanced Technology Laboratories, Schenectady, New York.

AF 04(611)-8020 "Analytical Techniques for Design of Static, Sliding and Rotating Seals for Use in Propulsion Subsystems," Illinois Institute of Technology Research Institute, Chicago, Illinois.

AF 04(611)-8176 "Development of Mechanical Fittings for Rocket Fluid Systems," Battelle Memorial Institute.

AF 04(611)-8392 "Seat and Poppet Development," Rocketdyne Division, North American Aviation, Canoga Park, California.

APPENDIX B

VALVE-PRESSURE-DROP CALCULATIONS

VALVE-PRESSURE-DROP CALCULATIONS

I. GENERAL DISCUSSION

A. Several valve types have been sized and the configuration sufficiently established to permit comparative analyses to be made. This appendix illustrates some of the computations and methods used to obtain pressure losses for the different designs. The approach is straightforward and unsophisticated because the designs are conceptual and lacking in detail necessary to conduct a rigorous analysis.

The valves considered as candidate components for this program and compared ΔP (pressure loss) are:

1. High-pressure in-line on-off or modulating sleeve valve.

2. High-pressure angle on-off or modulating sleeve valve.

3. Low-pressure ring-gate pump-suction valve integral with 180° flow reversal manifold for the oxidizer-pump inlet.

4. Low-pressure in-line on-off sleeve valve (4 required) combined with separate 180° flow reversal manifold at the oxidizer-pump inlet.

5. Low-pressure ring-gate fuel-pump suction valve, tank.

6. High-pressure in-line on-off venturi valve.

7. High-pressure in-line on-off butterfly valve.

8. High-pressure in-line on-off ball valve

9. High-pressure in-line rotary sleeve valve.

B. NOMENCLATURE AND UNITS

The following nomenclature is used throughout the calculations for each of the valves. The units of length, weight, volume, or time are given with each symbol or abbreviation as applicable.

Symbol	Definition	Unit
A	Valve inlet or line-flow area	in.^{2}
A_b	Blockage area of flow obstructions	in.^{2}
* C_D	Empirical drag-shape coefficient	ND
D	Valve inlet or line inside diameter	in.
** f	Friction factor for line-friction loss	ND
F_D	Drag force	lb
g	Acceleration of gravity	ft/sec^2
K	Empirical coefficient of kinetic energy loss	ND
L	Length of valve or line	in.
ΔP	Pressure loss, differential pressure	psi
psi	Unit pressure	lb/in.^{2}
R	Throat dia to line dia ratio	ND
V	Fluid velocity	ft/sec
$\dot{w}$	Fluid flow rate	lb/sec
ρ	Fluid density	lb/ft^3

* C_D values were otained from "Elementary Fluid Mechanics," by J. K. Vennard, third edition.

** Reynolds number is considered to be greater than 1×10^6 and the valves are assumed to have smooth flow passages with an ϵ/D ratio of 0.0002; therefore, f = 0.014 (Moody Diagram) will be used for all equivalent line-friction calculations.

II. Pressure-loss determination for an in-line sleeve-valve configuration is similar to that shown in Figure VII-C-1, as a function of diameter, flow rate, and fluid density.

A. The pressure loss through the valve is considered to be a summation of:

1. The sleeve-restriction loss at the valve inlet.

2. The friction loss of an equivalent length of straight pipe.

3. The drag losses incurred flowing past the support webs.

4. The drag losses incurred flowing around the central body.

B. The following assumptions are used:

1. The port configuration consists of 3/4-in.-wide slots around the circumference of diameter D. The slots are D/3 long and they are separated by 1/4-in.-wide ribs. The port area is $0.75 \times \pi D \times D/3 = 0.785\ D^2$ equal to the valve-inlet area.

2. The annular flow section, which encompasses the central body (sleeve and actuator), has an area of $0.785\ D^2$ equal to the valve-inlet area. The L/D of this section has been doubled for computing line-friction loss due to the larger-diameter annular configuration.

C. The calculated pressure losses are as follows:

1. Inlet-Sleeve Restriction ΔP_1.

The inside sleeve diameter in the inlet port is approximately 0.86D and is tapered to provide a convergent pipe section. The loss will be treated as a loss of a convergent pipe section.

$$AP_1 = \frac{0.02\ \rho V_2^2}{2g \times 144} \quad \text{psi}$$

$$V_2 = \frac{\dot{w}}{\rho A_2} \quad \text{ft/sec}, \qquad A_2 = \frac{0.785\ (.86D)^2}{144} = \frac{0.58D^2}{144}$$

$$V_2^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.336\ D^4} = \frac{6.18 \times 10^4\ \dot{w}^2}{\rho^2 D^4}$$

$$\Delta P_1 = \frac{0.02 \times \rho \times 6.18 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho^2 \times D^4}$$

$$\underline{\underline{\Delta P_1 = \frac{0.133\ \dot{w}^2}{\rho D^4} \quad \text{psi}}} \qquad \text{(Eq 1)}$$

2. Equivalent pipe-line friction loss, P_2.

$$\Delta P_2 = \frac{f\ L/D\ \rho V^2}{2g \times 144} \quad \text{psi}$$

$$V = \frac{\dot{w}}{\rho A} \quad \text{ft/sec} \qquad A = \frac{.785\ D^2}{144} \quad \text{ft}^2$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617\ D^4} = \frac{(3.36 \times 10^4)\ \dot{w}^2}{\rho^2 D^4}$$

$L = 3D$; therefore $L/D = 5$ (increased to account for double-walled annulus)

$f = 0.014$ (assumed)

$$\therefore \Delta P_2 = \frac{0.014 \times 5 \times \rho (3.36 \times 10^4)\ \dot{w}^2}{64.4 \times 144\ \rho^2 D^4}$$

$$\underline{\underline{\Delta P_2 = 0.255 \frac{\dot{w}^2}{\rho D^4} \quad \text{psi}}} \qquad \text{(Eq 2)}$$

3. Drag loss caused by port webs, ΔP_3

$$F_D = C_D A_b \frac{\rho V^2 \text{ lb}}{2 g \times 144}$$

$$\Delta P_3 = \frac{C_D A_b \rho V^2 \times \rho V}{2g \times 144 \dot{w}}$$

$$A_B = \frac{.25\pi D}{144} \times D/3 = \frac{0.262 D^2}{144} \quad ft^2$$

$$V = \frac{\dot{w}}{\rho A} \quad \text{ft/sec} \qquad A = \frac{0.785 D^2}{144} \quad ft^2 \text{ (by design)}$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617 D^4} = (3.36 \times 10^4) \frac{\dot{w}^2}{\rho^2 D^4}$$

C_D = 0.6 Empirical drag-shape coefficient.

$$\therefore \Delta P_3 = \frac{0.6 \times 0.262 D^2 \times \rho \times 3.36 \times 10^4 \times \dot{w}^2 \times \rho \times \dot{w} \times 144}{64.4 \times 144 \times \rho^2 \times D^4 \times 144 \times \dot{w} \times \rho \times .785 D^2}$$

$$\Delta P_3 = 0.725 \frac{\dot{w}^2}{\rho D^4} \quad \text{psi} \qquad \text{(Eq 3)}$$

4. Drag loss caused by central body P_4.

$$F_D = C_D \frac{A_b \rho V^2}{2 \times g \times 144}$$

$$\Delta P_4 = \frac{C_D A_b \rho V^2}{2 \times g} \times \frac{\rho V}{144 \dot{w}}$$

$$V = \frac{\dot{w}}{A} \text{ ft/sec} \qquad A = \frac{0.785\ D^2}{144} \text{ ft}^2 \text{ (design)}$$

$$V^2 = (3.36 \times 10^4)\ \frac{\dot{w}^2}{\rho^2 D^4}$$

$$A_B = \frac{.785(1.15D)^2}{144} = \frac{1.04D^2}{144} \text{ ft}^2$$

C_D = 0.15 Empirical shape-drag coefficient.

$$\Delta P_4 = \frac{0.15 \times 1.04\ D^2 \times \rho \times 3.36 \times 10^4 \times \dot{w}^2 \times \rho \times \dot{w} \times 144}{64.4 \times 144 \times \rho^2 \times D^4 \times 144 \times \dot{w} \times \rho \times 0.785\ D^2}$$

$$\Delta P_4 = 0.72\ \frac{\dot{w}^2}{\rho D^4} \text{ psi} \qquad \text{(Eq 4)}$$

5. Total pressure loss, ΔP.

The total pressure loss through the valve is the summation of pressure losses $\Delta P_1, \Delta P_2, \Delta P_3$, and ΔP_4.

$$\Delta P = \Sigma \Delta P_n = (0.33 + 0.255 + 0.725 + 0.72)\ \frac{\dot{w}^2}{\rho D^4}$$

Overall $$\Delta P = 1.83\ \frac{\dot{w}^2}{\rho D^4} \text{ psi} \qquad \text{(Eq 5)}$$

D. CONCLUSIONS

The previous analysis indicates that the major pressure losses occur where the fluid passes the webbed port section and changes direction because of central body section. Careful design and contouring of the webs and the ends of the central body section would result in reduced losses at these points.

III. Pressure-loss determination for an angle sleeve-valve configuration similar to that shown in Figure VII-C-2 as a function of diameter, flow rate, and fluid density.

A. The pressure loss through the valve is considered to be a summation of:

1. The sleeve-restriction loss at the valve inlet.

2. The friction loss of an equivalent length of straight pipe.

3. The directional-flow loss caused by the flow-diverter inefficiency.

4. The drag loss caused by the sleeve port-dividing webs.

5. The kinetic-energy loss in the torus collector.

B. The following assumptions and coefficients are used in the calculations:

1. The flow diverter is assumed to be 80% efficient.

2. The drag coefficient (C_D) for the port webs is assumed at 0.6. The port area consists of slots 3/4 in. by D/3 in. separated by 1/4-in. ribs. The port area is $0.75D \times \pi \times D/3 = .785\ D^2$ equal to the valve-inlet area.

3. 50% loss of kinetic energy is assumed in redirecting the radial flow through the torus collector.

C. The calculated pressure losses are as follows:

1. Inlet sleeve diameter is 0.86D and is tapered to offer a short contracting section. The loss will be treated as the loss of a convergent-pipe section.

$$\Delta P_1 = \frac{0.02\ \rho V_2^2}{2g \times 144} \quad \text{psi}$$

$$V_2 = \frac{\dot{w}}{\rho A_2} \quad \text{ft/sec} \qquad A_2 = \frac{0.785\ (.86D)^2}{144} = \frac{0.58D^2}{144}$$

$$V_2^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.336\ D^4} = \frac{6.18 \times 10^4\ \dot{w}^2}{\rho^2 D^4}$$

$$\Delta P_1 = \frac{0.02 \times \rho \times 6.18 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho^2 \times D^4}$$

$$\Delta P_1 = \frac{0.133\ \dot{w}^2}{\rho D^4} \quad \text{psi} \qquad \text{(Eq 1)}$$

2. Equivalent line-friction loss, ΔP_2

$$\Delta P_2 = \frac{f (L/D) \rho V^2}{2g \times 144} \quad \text{psi}$$

$$V = \frac{\dot{w}}{\rho A} \quad \text{ft/sec} \qquad A = \frac{.785 \, D^2}{144} \quad \text{ft}^2$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \, 0.617 \, D^4} = \frac{(3.36 \times 10^4) \, \dot{w}^2}{\rho^2 D^4}$$

$$f = 0.014 \text{ and } L/D = 5 \quad \text{(assumed)}$$

$$\Delta P_2 = \frac{0.014 \times 5 \times \rho \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 1.44 \times \rho^2 \, D^4}$$

$$\Delta P_2 = \frac{0.255 \, \dot{w}^2}{\rho D^4} \quad \text{psi} \qquad \text{(Eq 2)}$$

3. Flow-diverter loss, ΔP_3

The flow diverter at the assumed 80% efficiency would cause a 20% loss in kinetic energy.

$$\Delta P_3 = \frac{.20 \, \rho V^2}{2g \times 144} \quad \text{psi}$$

The velocity entering the diverter is that in the restricted sleeve diameter (0.86D)

$$V = \frac{\dot{w}}{\rho A} \quad \text{ft/sec} \qquad A = \frac{.785 \, (.86D)^2}{144} \quad \text{ft}^2$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.336\ D^4} = \frac{(6.18 \times 10^4)\ \dot{w}^2}{\rho^2\ D^4}$$

$$\Delta P_3 = \frac{.20 \times \rho \times 6.18 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho^2 \times D^4}$$

$$\Delta P_3 = \frac{1.33\ \dot{w}^2}{\rho D^4} \quad \text{psi} \qquad \text{(Eq 3)}$$

4. Drag loss caused by port webs, ΔP_4

$$F_D = C_D\ A_b\ \frac{\rho V^2}{2g \times 144} \quad \text{lb}$$

$$\Delta P_4 = \frac{C_D\ A_b\ \rho V^2}{2g \times 144} \times \frac{V}{\dot{w}} = \frac{C_D\ A_b^2\ \rho^2 V^3}{288\ g \times \dot{w}}$$

$$A_b = .25\ \frac{\pi D}{144} \times D/3 = \frac{0.262\ D^2}{144} \quad \text{ft}^2$$

$$V = \frac{\dot{w}}{\rho A} \ \text{ft/sec}, \quad A_{(\text{port area})} = \frac{.785\ D^2}{144} \ \text{ft}^2 \ \text{(by design)}$$

$$V^3 = \frac{\dot{w}^3 \times 144^3}{\rho^3 \times 0.484\ D^6} = \frac{(6.18 \times 10^6)\ \dot{w}^3}{\rho^3\ D^6}$$

$$C_D = 0.6 \ \text{Empirical drag-shape coefficient}$$

$$\Delta P_4 = \frac{0.6 \times 0.262\ D^2 \times \rho^2 \times 6.18 \times 10^6 \times \dot{w}^3}{288 \times 32.2 \times 144 \times \dot{w} \times \rho^3 \times D^6}$$

$$\Delta P_4 = \frac{0.728\ \dot{w}^3}{\rho D^4} \quad \text{psi} \qquad \text{(Eq 4)}$$

5. Torus collector loss, ΔP_5

$$\Delta P_5 = \frac{0.5 \rho V^2}{2g \times 144} \quad \text{psi}$$

$$V \text{ entering torus} = \frac{\dot{w} \times 144}{\rho \times 0.785 D^2}$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617 D^4} = \frac{3.36 \times 10^4 \times \dot{w}^2}{\rho^2 D^4}$$

$$P_5 = \frac{0.5 \times \quad \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho^2 \times D^4}$$

$$\underline{\underline{P_5 = 1.80 \frac{\dot{w}^2}{D^4} \quad \text{psi}}} \qquad \text{(Eq 5)}$$

6. Total valve-pressure loss, ΔP

The total pressure loss through the valve is the summation of the pressure losses ΔP_1, ΔP_2, ΔP_3, ΔP_4, and ΔP_5

$$\Delta P = \sum \Delta P_n = (0.133 + 0.255 + 1.33 + 0.728 + 1.80) \frac{\dot{w}^2}{\rho D^4}$$

$$\underline{\underline{\Delta P = 4.24 \frac{\dot{w}^2}{\rho D^4} \quad \text{psi}}} \qquad \text{(Eq 6)}$$

D. CONCLUSIONS

The coefficients and the efficiencies assumed for the various incremental sections of the valve are conservative; it is probable that effective contouring of the flow diverter and torus collector could reduce the P somewhat.

IV. Pressure-loss determination for the AJ-1 oxidizer pump-suction ring-gate valve, including the external elbows and manifolding to provide a 180-degree turn of the fluid flow at the pump-suction inlet.

A. The configuration of the valve is shown in Figure VII-E-1. The analysis includes the losses from a point just upstream of the suction-line transition elbow to the pump inlet after executing a 180° bend.

B. Assumptions and Conditions

1. The transition elbow executes a smooth 90° radius at the same time flaring out to distribute the flow into the torus collector. A loss coefficient, K = 0.3, is realistic for this part, based on Beij loss coefficients for smooth 90° bends.

2. Distribution of flow into the torus collector is such that 70% of the flow passes directly through the ring-gate port. The remaining 30% requires partial redirection. A conservative assumption would be that 30% of the flow loses all its kinetic energy. K = 0.3 will be used for this ΔP.

3. Two sets of supporting webs for the ring gate and flow diverter are contoured to present minimum flow resistance. The cross-sectional area of each set of webs exposed to the flow path is 24 in.^{2}. Drag-shape coefficients of 0.2 and 0.15 will be used to obtain the drag ΔP.

4. The flow diverter is roughly equivalent to a smooth radius 90° elbow. A factor K = 0.4 is assumed conservative for this ΔP.

5. This valve has been sized for the AJ-1 engine. For purposes of direct comparison with other valve arrangements the actual line and port areas, fluid-flow rates and fluid density, will be used to establish numerical Δ P valves.

Oxidizer flow rate = 13410 lb/sec (LO_2)

Fluid density = 70.5 lb/ft^3

Flow area (inlet) = 1133 $in.^2$

Flow area (outlet) = 1020 $in.^2$ (36 in.-dia pump inlet)

C. The incremental pressure losses are calculated for the inlet elbow, the drag losses at the port webs, and the redirection losses of the flow diverter.

1. Transition elbow pressure loss ΔP.

$$\Delta P_1 = \frac{K \times 2.24\, \dot{w}^2}{\rho A^2}$$

$K = 0.3$ Smooth radius 90° elbow ($r = 1.5$ D)

$$\Delta P_1 = \frac{0.3 \times 2.24 \times 13410^2}{70.5 \times 1133^2}$$

$$\Delta P_1 = \underline{\underline{1.33 \text{ psi}}}$$

2. Distribution torus loss ΔP_2.

$$\Delta P_2 = \frac{0.3 \times 2.24\ \dot{w}^2}{\rho A^2}$$

$$= \frac{0.3 \times 2.24 \times 13410^2}{70.5 \times 1133^2}$$

$$\Delta P_2 = \underline{\underline{1.33 \text{ psi}}}$$

3. Drag-pressure loss at webs ΔP_3.

$$P_3 = \frac{F_d}{A} \quad \text{psi}$$

$$F_D = \frac{C_D \, A_b \, V^2}{2g \times 144} \quad \text{lb}$$

$$A = 1133 \text{ in.}^2$$

$$A_{b1} = A_{b2} = 24 \text{ in.}^2, \quad C_{D1} = 0.2 \text{ upstream}, \quad C_{D2} = 0.15 \text{ downstream}$$

$$C_{D1}A_b + C_{D2}A_b = (C_{D1} + C_{D2})\,A_b = 0.35\,A_b = C_D A_b$$

$$V = \frac{\dot{w} \times 144}{\rho A} \text{ ft/sec} \qquad V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 A^2}$$

$$\Delta P_3 = \frac{0.35\,A_b \times \dot{w}^2 \times 144}{2g \times \rho \times A^3} = \frac{0.783 \times A_b \times \dot{w}^2}{\rho \times A^3}$$

$$= \frac{0.783 \times 24 \times (1.3410)^2 \times 10^8}{70.5 \times (1.133)^3 \times 10^9}$$

$$\Delta P_3 = \underline{\underline{0.033 \text{ psi}}}$$

4. 90° flow-director loss ΔP_4

$$\Delta P_4 = \frac{K \times 224\,\dot{w}^2}{\rho A^2}$$

$K = 0.4$ fairly smooth 90° Elbow (1020-in.^{2} exit area)

$$\Delta P_4 = \frac{0.4 \times 2.24 \times 13410^2}{70.5 \times 1020^2}$$

$$\Delta P_4 = \underline{\underline{2.17 \text{ psi}}}$$

5. Total valve and inlet ΔP

$$\Delta P = \Sigma \Delta P_n = 1.33 + 1.33 + 0.03 + 2.17$$

$$\text{Total } \Delta P = \underline{\underline{4.86 \text{ psi}}}$$

6. Converting the total loss to terms of pump inlet D for future use.

$$A \text{ outlet} = 1133 \text{ in.}^2 = 0.785 \ D^2$$

$$A \text{ inlet} = 1020 \text{ in.}^2 = 0.872 \ D^2$$

$$\Delta P_1 = \frac{0.3 \times 2.24 \ \dot{w}^2}{\rho \ 0.76 \ D^4} = \frac{.883 \ \dot{w}^3}{\rho \ D^4}$$

$$\Delta P_2 = \frac{.883 \ \dot{w}^2}{\rho \ D^4}$$

$$\Delta P_3 = \frac{0.783 \times 0.0212 \times \dot{w}^2}{\rho \times .617 \ D^4}, = \frac{0.0219 \ \dot{w}^2}{\rho \ D^4}$$

$$\Delta P_4 = \frac{0.4 \times 2.24 \ \dot{w}^2}{\rho \times .617 \ D^4} = \frac{1.45 \ \dot{w}^2}{\rho \ D^4}$$

$$\text{Total } \Delta P = \underline{\underline{\frac{3.22 \ \dot{w}^2}{\rho \ D^4}}} \text{ psi}$$

V. Pressure-loss determination of the AJ-1 oxidizer suction inline sleeve valve and manifold.

A. The configuration of the valve is similar to the high-pressure inline sleeve valve considered in Section II of this appendix (Figure VII-C-1). Analysis of this valve and the 180° return manifold at the oxidizer pump suction are presented for comparison with the ring-gate oxidizer pump-suction valve considered in Section IV. Four 20-in.-dia valves will be used, one in each of the oxidizer-pump feed lines. The lines downstream from the valves will discharge into a 180° bend manifold to redirect the fluid into the pump suction.

B. Assumptions and Conditions

1. The formula derived for the inline sleeve valve considered in Section II will be used to determine the valve pressure loss.

2. The 180° bend manifold closely approximates a close return 180° pipe bend. An empircal coefficient of kinetic energy loss, K = 1.2, will be used as a conservative value to determine this pressure loss.

3. Because this analysis is for direct comparison with the suction ring-gate valve considered in Section IV, the LO_2, oxidizer pressure, flow rate, and density from the AJ-1 engine specifications will be used.

$$\text{Total flow rate} = 13410 \text{ lb/sec}$$

$$\text{Flow per valve} = \dot{w} = \frac{13410}{4} \text{ lb/sec} = 3352 \text{ lb/sec}$$

$$\text{Fluid density } (LO_2) = \rho = 70.5 \text{ lb/ft}^3$$

C. The calculated pressure losses are as follows:

1. Valve-pressure loss ΔP_1

$$\Delta P_1 = \frac{1.88}{\rho} \frac{\dot{w}^2}{D} \text{ psi (Appendix A, Section II, C, 5)}$$

$$= \frac{1.88 \times 3352^2}{70.5 \times 20^4}$$

$$\Delta P_1 = \underline{\underline{1.84 \text{ psi}}}$$

2. Manifold (180° bend) pressure loss ΔP_2

$$\Delta P_2 = \frac{K\rho V^2}{2g \times 144}$$

$$K = 0.9 = \text{loss coefficient for smooth 180° bend } (r = 1.5D)$$

$$V = \frac{\dot{w} \times 144}{\rho A} \text{ ft/sec, } A = 0.785\ D^2 \text{ in.}^2$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617\ D^4}$$

$$\Delta P_2 = \frac{0.9 \times 144 \times \dot{w}^2}{64.4 \times 0.617 \times \rho \times D^4} = \frac{3.38\ \dot{w}^2}{\rho D^4}$$

For the AJ-1:

$$\Delta P_2 = \frac{3.38 \times 3352^2}{70.5 \times 20^4} = \underline{\underline{3.38 \text{ psi}}}$$

3. Total ΔP of valves and manifold

$$\text{Total } \Delta P = \Delta P_1 + \Delta P_2 = 1.84 + 3.38 = \underline{\underline{5.22 \text{ psi}}}$$

VI. Pressure-loss determination for low-pressure ring-gate fuel pump-suction valve designed for mounting in the fuel-tank outlet.

A. This valve is similar in configuration to the oxidizer pump-suction valve except that the tank itself replaces the collector torus. The configuration is such that the fluid enters the valve at an angle θ with the radial plane, which imparts an axial-velocity component, thereby reducing the pressure loss required to divert from radial to axial flow. A sectional view of the valve configuration is shown in Figure VII-E-1.

B. Assumptions

1. Valve-exit-port diameter = 29.5 in. equal to the pump-suction inlet.

2. The suction port is assumed to be unrestricted with rounded entry (C_D = 0.98). (Discharge coefficient of round-edged orifice.)

3. The valve design provides a full, open-valve height, h = 11 in.

4. The loss of the valve will be a summation of the following losses:

   a. Entrance-loss coefficient = 0.02 for well-rounded entry.

   b. Loss due to support webs having 22-in.^{2} normal area. Drag-shape coefficient = 0.20.

c. Loss due to redirection of radial flow. Assume flow diverter same as smooth 90° elbow, r = 1.5D. Beijs loss coefficient = 0.30.

d. Loss due to smooth convergence from valve-inlet area at h = 11 in. to valve outlet area D = 29.5 in. Loss coefficient = 0.04 based on exit velocity.

5. The model shown in Figure VII-E-1 is assumed for analysis of this valve.

C. Calculations for Valve ΔP (full open)

1. Entrance loss ΔP_1

$$\Delta P_1 = \frac{0.02 \ \rho V_e^2}{2g \times 144} \text{ psi}$$

$$V_e = \frac{\dot{w} \times 144}{\rho A_e} \text{ ft/sec} \qquad A_e = \frac{\pi}{2} \times \left[30.5 + (49.5 - 20 \sin G)\right] \times \left[\frac{h + 6 - 10}{\cos\theta}\right]$$

$$\theta = \tan^{-1} \frac{9.5}{h + 6} \quad \text{sleeve stroke} \quad h = 11 \text{ in. (design)}$$

$$\theta = \tan^{-1} \frac{9.5}{17} = \tan^{-1} .559 = 29.2°$$

$$\sin\theta = .488 \qquad \cos\theta = .873$$

$$A_e = \frac{\pi}{2} \times \left[30.5 + (49.5 - 9.76)\right] \times \left[19.5^{-10}\right] \text{ in.}^2$$

$$A_e = \frac{\pi \times 70.24 \times 9.5}{2} = 1047 \text{ in.}^2$$

Convert to terms of exit (pump suction) diameter,

Exit diameter $D = 29.5$ in.

A exit $= 0.785\ D^2 = 684$ in.^{2}

$$A_e = \frac{1047}{684} \times \frac{0.785\ D^2}{144} = 1.2\ D^2$$

$$V_e = \frac{\dot{w} \times 144}{\rho \times 1.2\ D^2}\ \text{ft/sec} \qquad V_e^{\,2} = \frac{(1.44 \times 10^4)\ \dot{w}^2}{\rho^2\ D^4}$$

$$\Delta P_1 = \frac{0.02 \times \rho \times 1.44 \times 10^4 \times \dot{w}^2}{64.4 \times 144\,\rho^2\ D^4}$$

$$\Delta P_1 = \frac{0.03\ \dot{w}^2}{\rho\ D^4}\ \text{psi} \qquad \text{(Eq 1)}$$

2. Drag loss due to support webs in flow path ΔP_2

$$\Delta P_2 = \frac{F_D}{A_e}\ \text{psi}$$

$$F_D = \frac{C_D\ A_b\ \rho\ V_e^{\,2}}{144 \times Zg}\ \text{lb}$$

$$V_e^{\,2} = \frac{1.44 \times 10^4\ \dot{w}^2}{\rho^2\ D^4}$$

$C_D = 0.20$ (assumed) $\quad A_b = 22$ in.^{2} (design)

$$F_D = \frac{0.20 \times 22 \times \rho \times 1.44 \times 10^4 \times \dot{w}^2}{1.44 \times 64.4 \times \rho^2\ D^4}$$

$$= 6.85\ \frac{\dot{w}^2}{\rho D^4}\ \text{lb}$$

$A_e = 1047 \text{ in.}^2$ (from VI, C, 1)

$$\Delta P_2 = \frac{6.85}{1047} \frac{\dot{w}^2}{\rho D^4}$$

$$\Delta P_2 = \frac{0.006 \ \dot{w}^2}{\rho D^4} \text{ psi}$$

3. Flow-diverter loss ΔP_3 (radial velocity only)

$$\Delta P_3 = \frac{0.3 \ \rho V_x^2}{29 \times 144} \text{ psi}$$

$V_x = V_e \cos \theta$ (radial component of velocity)

$$V_e = \frac{\dot{w} \times 144}{\rho \times 1.2 \ D^2} \quad \text{(from VI, C, 1)}$$

$$V_x = \frac{.873 \times \dot{w} \times 144}{\rho \times 1.2 \ D^2} \text{ ft/sec}$$

$$V_x^2 = \frac{(1.1 \times 10^4) \ \dot{w}^2}{\rho^2 D^4}$$

$$\Delta P_3 = \frac{0.3 \times \rho \times 1.1 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho^2 \times D^4}$$

$$\Delta P_3 = 0.356 \frac{\dot{w}^2}{\rho D^4} \text{ psi}$$

4. Smooth convergence loss ΔP_4

$$\Delta P_4 = \frac{0.04 \; \rho V^2_{exit}}{2g \times 144} \; \text{psi}$$

$$V_{exit} = \frac{\dot{w} \times 144}{\rho \, A_{exit}} \; \text{ft/sec} \qquad A_{exit} = .785 \, D^2 \; \text{in.}^2$$

$$V^2_{exit} = \frac{\dot{w} \times 144^2}{\rho^2 \times 0.617 \, D^4} = \frac{(3.36 \times 10^4) \, \dot{w}^2}{\rho^2 \, D^4}$$

$$\Delta P_4 = \frac{0.04 \times \rho \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho^2 \times D^4}$$

$$\Delta P_4 = 0.145 \; \frac{\dot{w}^2}{\rho \, D^4}$$

5. Total Valve Loss ΔP

$$\Delta P = \sum \Delta P_n = (0.03 + 0.006 + 0.356 + 0.145) \frac{\dot{w}^2}{\rho \, D^4}$$

$$\Delta P = 0.54 \; \frac{\dot{w}^2}{\rho \, D^4} \; \text{psi}$$

6. Used as the fuel (LH_2) pump-suction valve for the AJ-1 engine; the pressure loss is:

$\dot{w}$ = 2240 lb/sec

ρ = 4.8 lb/ft^3

D = 29.5 in.

$$\Delta P = \frac{0.54 \times (2240)^2}{4.8 \times (29.5)^4}$$

$$= \underline{\underline{0.745 \text{ psi}}}$$

VII. Pressure-loss determination for a high-pressure in-line venturi-type on-off valve having a configuration shown in Figure VII-C-10.

A. The pressure loss through the valve is considered to be a summation of:

1. The drag loss incurred by flow past the central obstruction formed by the poppet assembly.
2. The convergence loss from line area to throat area.
3. The divergence loss in the 10° diffuser section.
4. The friction loss of an equivalent length of straight pipe.

B. ASSUMPTIONS AND CONDITIONS

1. The configuration of the valve for analytical purposes is shown in Figure VII-C-10. The valve length and pressure-drop determination are dependent on the throat-to-line diameter ratio and the line diameter.

2. Valve size vs diameter- and area-ratio assumptions based on design*

a. R = throat dia/line dia, $D_t = R\ D$
b. Diffuser included angle = 10°
c. Poppet dia = 1.1 x throat dia
d. Poppet length = 3 x throat dia

*Data obtained from Mr. Z. Fox of Fox Valve Co.

3. Length Determination

$$L_D = \frac{D - D_T}{2 \tan 5°} = 5.7\ (D - D_T) = \underline{5.7D\ (1-R)}, \quad R = D_T/D$$

$$L_I = 3D_T = \underline{3RD}$$

$$L = L_D + L_I = 5.7\ D\ (1-R) + 3RD = \underline{\underline{2.7D\ (2.1-R)}}$$

4. The central poppet-assembly obstruction is assumed to be streamlined so that a drag coefficient, $C_D = 0.25$, will be realistic. The annular area is equal to the line area.

5. A loss coefficient, K = 0.04, is assumed for the rounded or bell-monthed gradual contraction to the throat.

6. A loss coefficient, K = 0.08, is assumed for the diffuser-section conical enlargement, based on empirical data from the Fox Valve Co.

7. The equivalent pipe length (L/D) assumed for friction loss is twice the physical length because of the annular inlet and the reduced throat and diffuser mean diameters.

C. The calculated pressure losses are as follows:

1. The drag loss caused by the central-body poppet assembly is:

$$\Delta P_1 = \frac{C_D \times A_b \times \rho V^2}{2g \times 144 \times A} \quad \text{psi}$$

The valve-inlet diameter is 1.2D and the annular area is equal to the line area $A = 0.785D^2$. Therefore the blocked area is by design:

$$A_b = .785\ (1.2D)^2 - (D)^2 = 0.346\ D^2$$

$$V^2 = \frac{\dot{w} \times 144}{\rho A}\ \text{ft/sec}, \qquad A = 0.785\ D^2$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617\ D^4} = \frac{3.36 \times 10^4\ \dot{w}^2}{\rho^2 \times D^4}$$

$$C_D = 0.25\ \text{(assumed)}$$

$$\Delta P_1 = \frac{0.25 \times 0.346\ D^2 \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 1.44 \times 0.785D^2 \times \rho \times D^4}$$

$$\underline{\underline{\Delta P_1 = 0.40\ \frac{\dot{w}}{\rho D^4}\ \text{psi}}} \qquad \text{(Eq 1)}$$

2. <u>Gradual-contraction loss</u> ΔP_2

$$\Delta P_2 = \frac{K \times V_T^2}{2g \times 144}\ \text{psi}$$

$K = 0.04$ for rounded entry contraction

$$V_T = \frac{\dot{w} \times 144}{\rho A_T} \qquad A_T = 0.785\ D_T^{\ 2}$$

$$D_T = RD \qquad A_T = 0.785\ R^2\ D^2$$

$$V_T^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617\ R^4 D^4} = \frac{3.36 \times 10^4 \times \dot{w}^2}{R^4 \rho^2\ D^4}$$

$$\Delta P_2 = \frac{0.04 \times 3.36 \times 10^4 \times w^2}{64.4 \times 144 \times R^4 \times \rho \times D^4}$$

$$\underline{\underline{\Delta P_2 = \frac{.145\ \dot{w}^2}{R^4 \times \rho D^4}\ \text{psi}}} \qquad \text{(Eq 2)}$$

3. Diffuser conical-expansion loss, ΔP_3

$$\Delta P_3 = K \frac{(V_T^2 - V^2)}{2g \times 144}\ \text{psi}$$

K = 0.08 empirical factor (Fox data)

$$V_T^2 = \frac{3.36 \times 10^4 \times \dot{w}^2}{R^4\ \rho^2\ D^4} \qquad \text{(see VII, C, 2)}$$

$$V^2 = \frac{3.36 \times 10^4 \times \dot{w}^2}{\rho^2\ D^4} \qquad \text{(see VII, C, 1)}$$

$$V_T^2 - V^2 = \frac{3.36 \times 10^4}{R^4} \left[(1 - R^4)\ \frac{\dot{w}^2}{\rho^2 D^4}\right]$$

$$\Delta P_3 = \frac{0.08 \times 3.36 \times 10^4\ (1-R^4)\ \dot{w}^2}{64.4 \times 144 \times R^4 \times \rho \times D^4}$$

$$\underline{\underline{\Delta P_3 = \frac{0.29}{R^4}\ (1 - R^4)\ \frac{\dot{w}^2}{\rho\ D^4}\ \text{psi}}} \qquad \text{(Eq 3)}$$

4. Equivalent pipe-friction loss ΔP_4

$$\Delta P_4 = \frac{f\ L/D\ \rho V^2}{2g \times 144} \text{ psi}$$

f = 0.014 (see I, B)

L/D = 2 x 2.7 (2.1 - R) (use 9 as average)

$$V^2 = \frac{3.36 \times 10^4 \times \dot{w}^2}{\rho^2\ D^4} \qquad \text{(see VII, C, 1)}$$

$$\Delta P_4 = \frac{0.014 \times 9 \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho \times D^4}$$

$$\Delta P_4 = 0.456\ \frac{\dot{w}^2}{\rho D^4} \text{ psi} \qquad \text{(Eq 4)}$$

5. Total valve loss ΔP

$$\Delta P = \Sigma P_n = \left[0.40 + \frac{0.145}{R^4} + \frac{0.29}{R^4}\ (1 - R^4) + 0.456\right] \frac{\dot{w}^2}{\rho D^4}$$

$$\Delta P = 0.435\ \left(1.30 + \frac{1}{R^4}\right) \frac{\dot{w}^2}{\rho D^4}$$

Converting to various D_T/D ratios

R	ΔP
0.8	$\Delta P = 1.89\ \frac{\dot{w}^2}{P_D{}^4}$
0.6	$\Delta P = 3.92\ \frac{\dot{w}^2}{P_D{}^4}$

R	ΔP
0.5	$\Delta P = 7.53 \frac{\dot{w}^2}{P_D^4}$
0.4	$\Delta P = 17.6 \frac{\dot{w}^2}{P_D^4}$
0.3	$\Delta P = 54.3 \frac{\dot{w}^2}{P_D^4}$
0.2	$\Delta P = 272 \frac{\dot{w}^2}{P_D^4}$ psi

VIII. BUTTERFLY VALVE, ΔP VERSUS SIZE, SYSTEM PRESSURE AND FLUID DENSITY

The loss in kinetic-energy pressure through the butterfly valve will be assumed to be the net change in flow kinetic energy occurring past the blade. This assumption is based on the fact that no recovery of the increased kinetic energy occurs downstream of the blade, as there is no gradual diffuser section in this location.

Line flow area upstream and downstream from the valve is

$$A_L = .785\ D^2$$

Net flow area past the valve blade is

$$A_U = .785\ D^2 - D \times (d + 1.25\ t), \text{ where } d = \text{shaft dia} = .006\ D \times P$$

$$t = 3.5 \times 10^3 \times D/S$$

(from size and weight analysis)

$$A_U = .785\ D^2 - D\ (.006\ D \times P + 3.5 \times 10^3\ D/S) = .785\ D^2 - .006\ D^2 \times P - .0044\ D^2/S$$

$$A_U = D^2\ (.785 - .006\ P - \frac{4400}{S})$$

Change in kinetic energy in passing the blade is

$$\Delta KE = \frac{2.24\ \dot{w}^2}{P}\ (\frac{1}{A_V^2} - \frac{1}{A_L^2}) = \frac{2.24\ \dot{w}^2}{P_D^4} \left[\frac{1}{(.785 - .006\ P - \frac{4400}{S})^2} - \frac{1}{.616}\right]$$

$$\Delta KE = \Delta P = \frac{2.24\ \dot{w}^2}{D^4} \left[\frac{1}{(.785 - .006\ P - \frac{4400}{S})^2} - 1.62\right]$$

NOTE: The number within the bracket in the above equation is the fraction of the kinetic energy of the flow at the line velocity that is lost through the valve. For a 1500-psi valve, blade stress = 35,000 psi, this factor is 0.73, which closely corresponds to the loss measured in water-flow tests of this design (Titan) valve.

$$\text{pressure loss } (\Delta P) = \underline{\underline{\left[\frac{2.24}{(.785 - .006\ P - \frac{4400}{S})} - 3.63\right] \frac{\dot{w}^2}{\rho D^4}\ \text{psi}}}$$

where:

ΔP = pressure loss, psi

$\dot{w}$ = flowrate, lb/sec

ρ = fluid density, lb/ft^3

D = line size (ID), in.

P = valve proof pressure, psi

S = Blade stress, psi

NOTE: Bearing stress is 15,000 psi with no shaft deflection.

IX. BALL-VALVE PRESSURE-LOSS DETERMINATION

It may be seen (Figure VII-C-20) that the only flow restriction is caused by the two circumferential grooves between the ball (moves) and the body. By careful design to minimize the width of these grooves, the pressure loss can be made nearly the same as an equal length of straight line. In practice, the loss has been found to be about 5 to 20% of the flow kinetic energy, or

$$\Delta P = .1 \times \frac{3.64\ \dot{w}^2}{\rho D^4} \simeq \underline{\underline{\frac{.364\ \dot{w}^2}{\rho\ D^4}\ \text{psi}}}$$

ΔP = Loss, psi

$\dot{w}$ = Flowrate, lb/sec

D = Line ID = Ball port dia, in. (Note: Some ball valves are made with port dia less than line ID to reduce envelope size.)

ρ = Fluid Density, lb/ft^3

X. ROTARY SLEEVE-VALVE PRESSURE-LOSS DETERMINATION

The pressure loss through the valve will be estimated by dividing the valve into several increments and calculating the loss caused by each. The sum of these increments is the total pressure loss. See Figure VII-C-13.

$$\text{TOTAL } \Delta P = \underline{\underline{\frac{4.96\ \dot{w}^2}{\rho D^4}}}\ \text{psi}$$

Where: $\dot{w}$ = Flow rate, lb/sec

ρ = Fluid Density, lb/ft^3

D = Valve inlet dia, in.

ΔP = Pressure loss, psi

A. Pressure loss due to conical flow diverter. The configuration simulates a smooth 90° elbow with $r = 1.5D$ for which a loss coefficient,

$K = 0.3$, is realistic

$$\Delta P_1 = \frac{0.3\ PV^2}{2g \times 144}$$

$$*\Delta P_1 = \frac{0.3 \times 3.64\ \dot{w}^2}{\rho D^4}$$

$$\Delta P_1 = \frac{1.09\ \dot{w}^2}{\rho D^4}\ \text{psi}$$

B. Loss due to flow through radial ports in rotating-sleeve section. This loss will be conservatively considered as flow through four sharp-edged orifices having the combined area $A_2 = 0.45\ D \times \frac{D}{2} = .717D^2$. A loss coefficient, $K = 0.8$, is assumed for this section

$$\Delta P_2 = \frac{0.8\ \rho V_2^{\ 2}}{2g \times 144}$$

$$V_2 = \frac{\dot{w} \times 144}{\rho A_2}\ \text{ft/sec} \qquad A_2 = .717D^2\ \text{in.}^2$$

$$V_2^{\ 2} = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.5 \times D^4} = \frac{4.15 \times 10^4\ \dot{w}^2}{\rho^2\ D^4}$$

$$*\text{Fluid KE} = \frac{V^2}{2g \cdot 144} = \frac{2.24\ \dot{w}^2}{\rho A^2}\ \text{psi} = \frac{3.64\ w^2}{\rho D^4}\ \text{psi}$$

$$\Delta P_2 = \frac{0.8 \times 4.15 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho \times D^4}$$

$$\Delta P_2 = \frac{3.58\ \dot{w}^2}{\rho\ D^4} \quad \text{psi}$$

C. Loss due to bend in the outer housing (axial). The kinetic-energy loss factor for a smooth radius, 90° elbow is 0.3 x KE. Assuming the outer housing to be similar to an elbow,

$$\Delta P_3 = 0.3 \times \frac{3.64\ w^2}{\rho D^4} = \frac{1.09\ \dot{w}^2}{\rho D^4} \quad \text{psi}$$

D. Friction loss for equivalent length of straight pipe. L/D = 4 is assumed for this configuration, which has a physical length of 2D (because of the annular flow passage)

$$\Delta P_4 = \frac{0.014 \times 4 \times V^2 \times \rho}{2g \times 144}$$

$$= \frac{0.056 \times 3.64\ \dot{w}^2}{\rho D^4}$$

$$\Delta P_4 = \frac{0.204\ \dot{w}^2}{\rho D^4} \quad \text{psi}$$

Total rotary sleeve-valve pressure loss is the sum of the incremental losses calculated above, $\Sigma \Delta P = (1.09 + 3.58 + 1.09 + 0.20)\ \frac{\dot{w}^2}{\rho D^4}$

$$\text{TOTAL} \quad \Delta P = \frac{4.96\ \dot{w}^2}{\rho\ D^4} \quad \text{psi}$$

XI. Pressure-loss determination for multiple-venturi integral pump-discharge valve.

A. The configuration of this valve is shown in Figure VII-D-3. The design is such that the annular flow passage around the poppet and actuator insert has a constant area throughout the length of the valve equal to the area entering the diffuser section (which diameter forms the poppet seat).

B. ASSUMPTIONS

1. The length of the contoured poppet assembly is three times the poppet-seat diameter.

2. The contoured poppet assembly will be considered as a central body blocking the flow path having an area $A_b = 1.2A$, where A is the area of the flow passage. The drag-loss coefficient, C_D, for this will be assumed as 0.08.

3. The longitudinal struts supporting the central body have a total area $A_b = 0.1A$. A drag-loss coefficient, $C_D = 0.06$, will be assumed for these thin struts.

4. A valve sized for one of eight separate pump-diffuser outlets will be based on the following parameters for the AJ-1 fuel pump.

Diffuser throat area, $A = 6.74 \text{ in.}^2$
Equivalent throat dia, $D = 2.92$ in.
Total flow rate of LH_2, $\dot{w} = 2240$ lb/sec
Flow rate/valve, $\dot{w} = 2240/8 = 280$ lb/sec
Density of LH_2, $\rho = 5.3 \text{ lb/ft}^3$

C. The calculated loss will be considered to be the sum of an equivalent line loss, drag loss of a central body, and drag loss of the supporting struts. The effects of the valve on the diffuser efficiency have not been determined and have not been included as part of the valve pressure loss.

1. Equivalent line-friction loss, ΔP.

$$\Delta P_1 = \frac{f \times L/D \times \rho V^2}{2g \times 144} \text{ psi}$$

$L/D = 6$ assumed for annular passage, $f = .014$ = friction factor.

$$V = \frac{\dot{w} \times 144}{\rho A} \text{ ft/sec}, \quad A = .785D^2 \text{ in.}^2$$

$$V^2 = \frac{\dot{w}^2 \times 144^2}{\rho^2 \times 0.617\ D^4} = \frac{3.36 \times 10^4\ \dot{w}^2}{\rho^2\ D^4}$$

$$\Delta P_1 = \frac{0.014 \times 6 \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times P \times D^4}$$

$$\Delta P = .304 \frac{\dot{w}^2}{\rho D^4} \text{ psi}$$

2. Drag loss due to central body, ΔP_2

$$\Delta P_2 = \frac{C_D A_b P V^2}{2g \times 144 \times A} \text{ psi}$$

$C_D = 0.08$ drag-shape coefficient

A_b = 1.2A, area of central body (design)

$$V^2 = \frac{3.36 \times 10^4 \dot{w}^2}{\rho^2 D^4} \qquad \text{(from XI, C, 1)}$$

$$\Delta P_2 = \frac{0.08 \times 1.2 \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho \times D^4}$$

$$\Delta P_2 = 0.348 \frac{\dot{w}^2}{\rho D^4} \text{ psi}$$

3. Drag loss due to support struts, ΔP_3

$$\Delta P_3 = \frac{C_D A_b P V^2}{2g \times 144 \times A}$$

C_D = 0.06 drag-shape coefficient

A_D = 0.1 A area of struts (design)

$$\Delta P_2 = \frac{0.06 \times 0.1 \times 3.36 \times 10^4 \times \dot{w}^2}{64.4 \times 144 \times \rho \times D^4}$$

$$\Delta P_3 = 0.022 \frac{\dot{w}^2}{\rho D^4} \text{ psi}$$

4. Total valve loss, ΔP

$$\Delta P = \Sigma \Delta P_n = (0.304 + 0.348 + 0.022) \frac{\dot{w}^2}{\rho D^4}$$

$$\Delta P = 0.674 \frac{\dot{w}^2}{\rho D^4} \text{ psi}$$

5. Based on the AJ-1 fuel-pump data:

$$\Delta P = \frac{0.674\ (280)^2}{5.3\ (2.93)^4}$$

$$\Delta P = \underline{\underline{135\ \text{psi}}}$$

Based on the total pump-outlet head of 4700 psi, this loss represents less than a 3% loss for a shutoff valve having excellent control capability as well as a very attractive small envelope and weight characteristics.

APPENDIX C

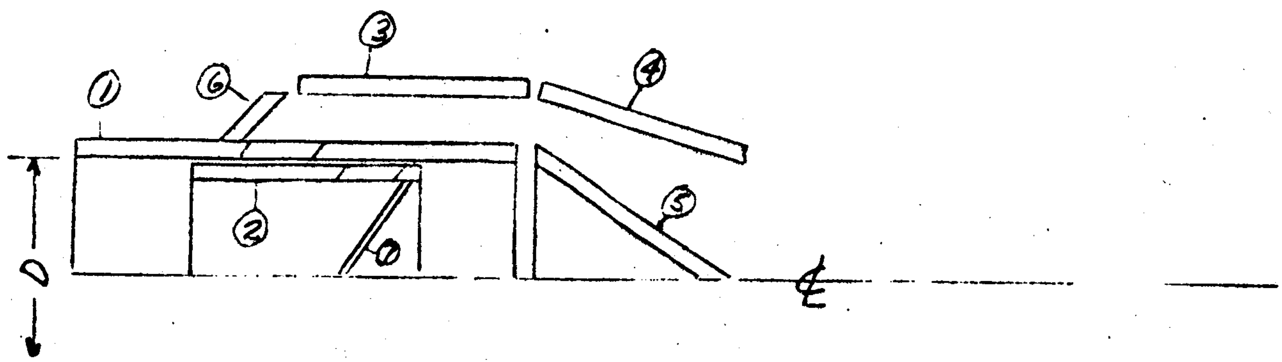
VALVE SIZE AND WEIGHT CONSIDERATIONS

VALVE SIZE AND WEIGHT CALCULATIONS

Size and weight studies have been made for most of the discrete valve types discussed in this report. This appendix includes some of the calculations used to establish the valve sizes and weights relative to line size, D, proof pressure, P, material allowable stress, S, and material density, ρ .

I. WEIGHT AS A FUNCTION OF SIZE, PRESSURE, AND MATERIAL FOR THE IN-LINE SLEEVE VALVE

The in-line sleeve valve may be represented as several cylinders and cones for the weight analysis as shown below:



By assuming that all dimensions are proportional to the inlet diameter, D, and that all thicknesses are proportional to the product of the pressure and the diameter divided by the strength of the material, the relationship of weight as a function of diameter, material strength, pressure, and material density can be derived as follows:

Part 1

$d_{ave} = 1.1D$; length = $l \doteq 2.5D$; t = thickness = $\frac{PD}{2S}$, S = stress.

Volume of material = $\pi(1.10) \times 2.5D \times PD/2S = \frac{4.3PD^3}{S}$. Weight of material = $\rho \times$ volume = $\frac{4.3\ \rho PD^3}{S}$

Part 2

$$d_{ave} = 0.9D, \quad l = 1.8D, \quad t = \frac{P(0.8D)}{2S},$$

$$\text{Material Weight} = \rho \times \pi d \times l \times t = \frac{2\ PD^3}{3}$$

Part 3

$$d_{ave} = 1.5D, \quad l = 1.5D, \quad t = \frac{P(1.5D)}{2S}$$

$$\text{Weight} = \rho \times \text{Volume} = \frac{5.3\ PD^3\rho}{S}$$

Part 4

$$d_{ave} = 1.3D, \quad l = D, \quad t = \frac{P(1.3D)}{2S}$$

$$\text{Weight} = \rho \times \text{Volume} = \frac{2.7\ PD^3\rho}{S}$$

Part 5

$$d_{ave} = 0.6D, \quad l = 0.8D, \quad t = \frac{P(0.6D)}{2S}$$

$$\text{Weight} = \frac{0.5\ PD^3\rho}{S}$$

Part 6

$$d_{ave} = 1.35D, \quad l = 0.3D, \quad t = \frac{P(1.35D)}{2S}$$

$$\text{Weight} = \frac{0.8\ PD^3\rho}{S}$$

Part 7

$A_{surface} = (0.785\ D^2)1.3, \quad t = 0.02D$ (not stressed)

$$\text{Weight} = \rho At = \frac{0.02\ D^3\rho}{S}$$

The total valve weight without actuator is the sum of the weights of the parts:

$$\text{Total Weight} = \sum(w_1 + W_2 + \ldots\ldots + W_7)$$

$$\text{Total Weight} = \frac{15.62\ PD^3\rho}{S}$$

Weight (W) = valve weight without actuator, lb

P = maximum design pressure (i.e., proof pressure), psi

D = line size (diameter), in.

ρ = material density, lb/in.^{3}

S = design stress limit (occurs at P), psi

II. ROTATING SLEEVE-VALVE WEIGHT (W/O ACTUATOR) VS SIZE, PRESSURE, AND MATERIAL

The weight of this valve will be determined by calculating the required thickness of the various parts with regard to size and strength. The other dimensions of the parts will be assumed proportional to the line ID. With the dimensions known, the volume of metal can be calculated, which multiplied by the material density yields the weight of the part.

Let: P = Proof Pressure, psig   D = Line Size, ID, in.

S = Material Stress, psi ρ = Material Density, lb/in.^{3}

t = Material Thickness

A. CENTRAL FLOW DIVERTER

Assume that a maximum flow rate will have a kinetic-energy pressure of 100 psi $(= \rho \frac{V^2}{2g \cdot 144})$.

Force on the Deflector = Δ Flow Momentum/time

$$\Delta \text{ Flow Momentum/Time} = \dot{M}\Delta V_x = \frac{\dot{W}}{g}\Delta V_x, \quad \dot{W} = \rho \frac{AV_x}{144},$$

Then:

$$\frac{\Delta M}{\text{Unit Time}} = \rho \frac{AV_x \Delta V_x}{144\ g} \qquad \Delta V_x = V_x - V_x \cos\theta \qquad (\theta = \text{Bend Angle of Flow})$$

$$\Delta V_x = V_x\ (\sec\theta - 1), \qquad \Delta M = \rho \frac{AV_x^2\ \ (\sec\theta - 1)}{144\ g}$$

$$\text{Since} \quad KE = \rho \frac{V^2}{2g \cdot 144}\ \text{psi}, \qquad \frac{\Delta M}{\Delta T} = KE \left(\frac{A(\sec\theta - 1)}{2} \right) = \text{Force on Deflector}$$

Assuming $P_{KE} = 100$ psi,

Because the average pressure on the deflector (to produce F) is $P_{av} = F/A$, $P_{av} = P_{KE} \frac{(\sec\theta - 1)}{2}$, with $P_{KE} = 100$ psi, $P_{AV} = 100 \frac{(\sec\theta - 1)}{2}$ $50(\sec\theta - 1)$psi, for this value, $\theta \approx 45°$, $\sec\theta = \sec 45° = 1.41$, $P_{av} = 50(1.41 - 1) =$ 21 psi (av) due to turning the flow, not static pressure.

Because the AV pressure resulting from turning the flow is small, this criteria will not be used to establish the diverter thickness. Although the diverter does not "see" the static pressure of the system, the occurrence of water hammer could cause an appreciable ΔP across this part, depending on the compressibility of the flow medium and the size of the passage whereby the pressure on either side of the diverter is equalized. Assuming the safety factor applied to the system pressure to account for water hammer is 2.5, the equivalent factor for the flow diverter is 2.5-1 = 1.5, because system pressure is initially equalized.

Based on 1.5 times system pressure on the upstream side of the diverter, the maximum pressure it must be designed to withstand is

$$P = 1.5 \times (\text{System Pressure}) = \frac{\text{Proof Pressure}}{2.5 \times 1.2} \times 1.5 = .5\ (\text{Proof Pressure})$$

Where Proof Pressure = System Pressure x 1.2 x 2.5 = 3 (System Pressure)

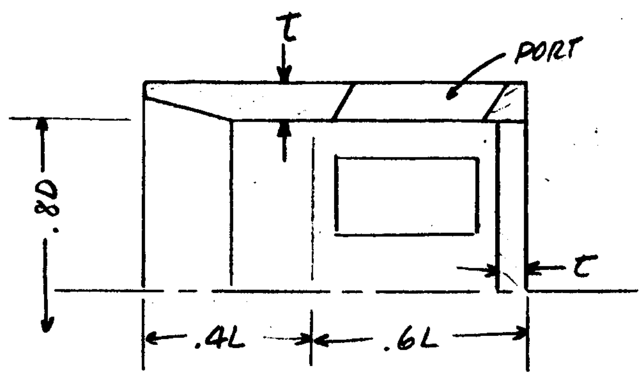
$$\text{Shear Area} = \pi D t, \quad \text{Stress} = \frac{.5P\,(.785D^2)}{\pi D t}$$

$$t = \frac{.5P(\pi/4)\,D^2}{\pi DS} = \frac{.125\ PD}{S}, \quad \text{Vol} = A \times t \approx 1.2D^2 \times \frac{.125\ PD}{S} = \frac{.15\ PD^3}{S}\ \text{in.}^3$$

$$\text{Wt of Flow Deflector} = \rho \times \text{Vol} = \underline{\underline{\frac{.15\,\rho PD^3}{S}\ \text{lb}}}$$

B. ROTATING SLEEVE

The primary concern in designing the sleeve is to limit diametral expansion (i.e., distortion) from the open- to closed-valve positions. This tolerance (for pressure and temperature distortion) is limited by the seal-clearance accommodation, which may be estimated. A 4-in. valve seal could probably accommodate a radial clearance change of .004 to .008 diametrical change. This amounts to $\frac{.008}{4}$ = 0.2% of the diameter (set S = .002E), where E = modulus of elasticity if required by seal criteria.



Pressure force acts on $\sim .6L = (.8D \times .6L)P = .5\ DLP$ lb

Approximately $.4L$ must carry the Hoop Stress $A = .4L \times 2t$

$$\text{Stress} = S = \frac{.5\ DLP}{.8Lt} = .625\ DP/t, \qquad t = \frac{.625\ DP}{S}$$

Area $\simeq .9\pi D \times .8L + .785(.8D)^2$.

For Port Area = Line Area: $.4\pi(.8D) \times .5L = .785(.8D)^2$

$L = 1 \times D$

Hence area $\simeq .9\pi D \times .8D + .5D = 2.26D^2 + .5D \simeq 2.46D^2$

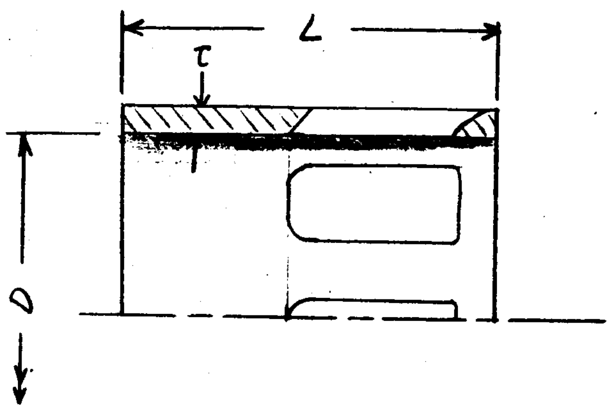
$$\text{Vol} = A \cdot T \simeq 2.46D^2 \times .625D\ P/S = \frac{1.54\ PD^3}{S}$$

$$\text{Sleeve wt} \quad \frac{1.54\,\rho PD^3}{S^*}$$

---

*S = .002E if required

C. INNER BODY, SLEEVE HOUSING



Assume L = 1.5D

Applying the same thickness as for the sleeve, ratioed to the diameter increase:

$$t \simeq 1.2\,(t_{sleeve}) = \frac{.75\ DP}{S}$$

$A = 1.05D \times 1.5D = 5D^2$, Vol $= A.t = 3.75D^3P/S$

Part wt $= \dfrac{3.75\,\rho\,PD^3}{S}$

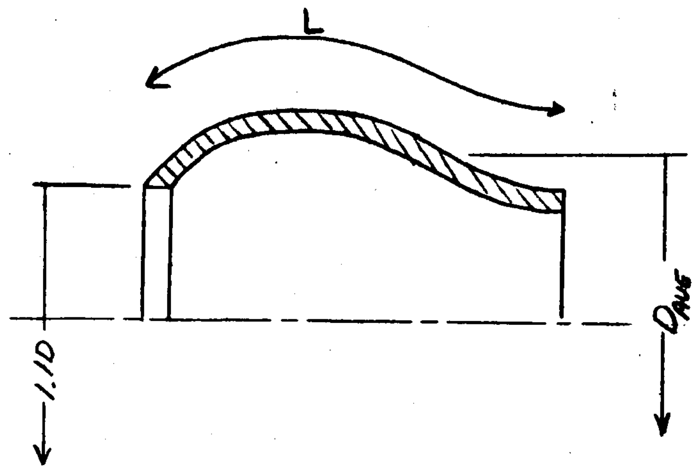
Inner-body Fairing Cone

Assume av $t = 2/3\ t_{body}$, $t = \dfrac{0.5\ DP}{S}$

$A = D/2 \times D = D^2/2$, Vol $= A.t = 0.8D^3P/S$

Part wt $= \dfrac{0.8\,\rho\,PD^3}{S}$

D. OUTER BODY



Average dia = 1.3D,

$$av\ t = \frac{PD_{av}}{2S} = \frac{1.3PD}{2S} = \frac{.65PD}{S}$$

$$L \approx 1.6D, \quad A = D_{av} \times L = 6.5D\ in.^2$$

$$Vol = A.t = \frac{4.22PD^3}{S}, \quad Part\ wt = \underline{\underline{\frac{4.22\,\rho PD^3}{S}\ lb}}$$

E. TOTAL VALVE WEIGHT WITHOUT ACTUATOR

$$Total\ wt = \sum Part\ wt = \underline{\underline{\frac{10.46\,\rho PD^3}{S}\ lb}}$$

Where: ρ = Material Density, lb/in.^{3}

P = Proof Pressure

S*= Material Stress at P = Proof Pressure (i.e., Design Stress)

D = Valve Inlet ID (i.e., Line Size)

---

* Set S = .002E to limit distortion to .2% if required by seals, where E = Material Modulus of Elasticity.

III. VENTURI TYPE VALVE--SIZE AND WEIGHT DETERMINATION

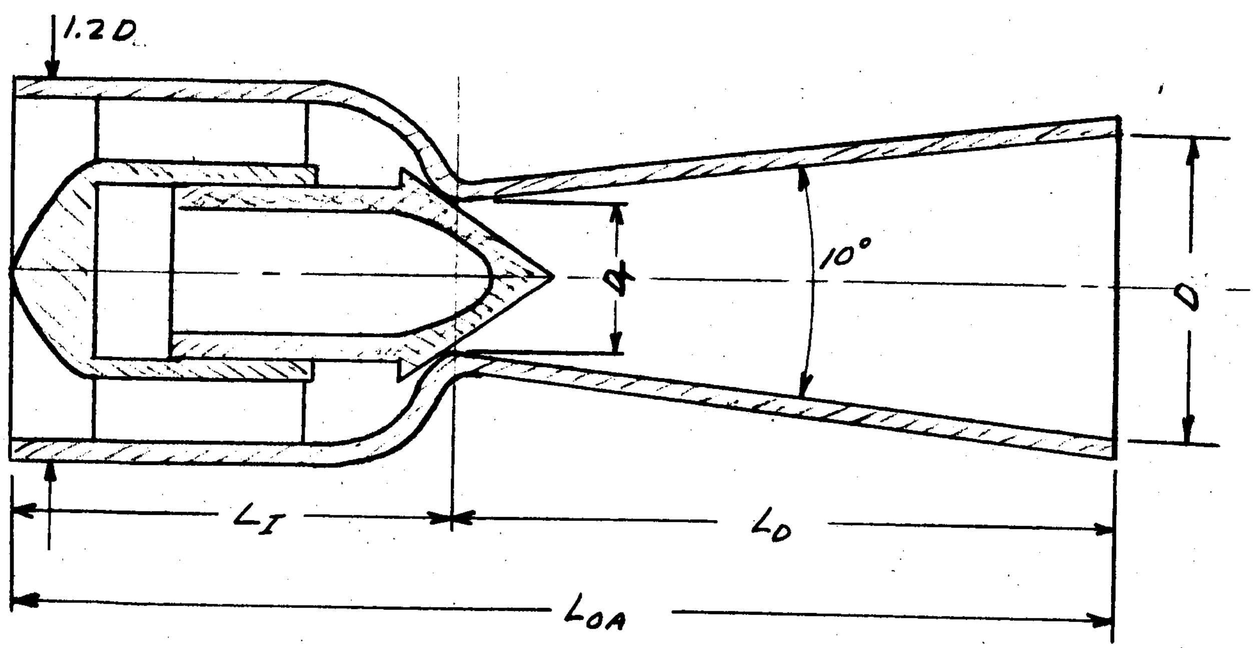
Valve size vs dia and area ratio.

Assume: R = throat dia/line dia, $D_T = R \cdot D$

Diffuser included angle = 10°
Poppet dia = 1.1 x throat dia
Poppet length = 3 x throat dia

Data obtained from Mr. Z. Fox

Sketch of Valve Layout



$$L_D = \frac{D - D_T}{2 \tan 5°} = 5.7(D_L - D_T) = \underline{5.7(1 - R)}, \quad R = D_T/D$$

$$L_I = 3D_T = \underline{3RD}$$

$$L_{OA} = L_D + L_I = 5.7D(1 - R) + 3RD = \underline{\underline{2.7D(2.1 - R)}}$$

A. WT OF DIFFUSER SECTION

$$\text{Surface area} \approx \frac{\pi}{2}(D + D_T) \times \frac{LD}{\cos 5^\circ} = 1.6(D + D_T)L_D$$

$$= 1.6D\,(1 + R) \times 5.7D_L\,(1 - R) = \underline{9.12D^2\,(1 - R^2)}$$

Assume uniform wall thickness throughout the outer shell:

$$t = \frac{PD}{2S}\ ,\ \therefore\ \text{Vol} = A \cdot t = \frac{9.12PD^3}{2S}(1 - R^2) = \frac{4.56PD^3}{S}(1 - R^2)$$

$$\text{Wt} = \rho \times \text{Vol} = \underline{\underline{\frac{4.56\,\rho PD^3}{S}(1 - R^2)}} \qquad \text{(Diffuser Only)}$$

B. WT OF INLET SECTION

Dia of inlet section = $D\sqrt{1 + 1.21R^2}$ at entrance, for flow area to equal line area. In practice, the flow area is reduced gradually upstream of the throat, which allows the assumption: inlet-section average dia $\approx$ 1.2 x line dia.

$$\text{Surface area of inlet section} = 1.2\pi D\,(3RD) = 11.3RD^2$$

$$t = \frac{P \times 1.2D}{2S},\ \text{Vol} = A \cdot t = \frac{6.8PD^3R}{S}\ ,\quad \text{wt} = \rho \times \text{Vol}$$

$$\underline{\underline{\text{Wt} = \frac{6.8\,\rho PRD^3}{S}}} \qquad \text{(Inlet Section Only)}$$

C. WT OF POPPET ASSEMBLY

AV poppet-assembly dia $\doteq 1.1D_T = 1.1RD$

Poppet-assembly length $\doteq 3D_T = 3RD$

Assume poppet-assembly compressive hoop stress = $\frac{1}{2}$ the outer body tensile hoop stress

$$t = \frac{PD_P}{\frac{1}{2}\cdot 2S} = \frac{PD_P}{S}, \quad Vol = \pi Dlt = \pi 1.1RD \times 3RD \times \frac{P(1.1RD)}{S}$$

$$Vol = \frac{11.4PR^3D^3}{S}, \quad Wt = \rho \times Vol \quad Wt = \frac{11.4\,\rho PR^3D^3}{S} \quad \text{(Cylinder Parts)}$$

The ends of the poppet assembly are circular; therefore, their weight is approximately

$$\text{Ends:} \quad A = 2(1.5)\frac{\pi}{4}D_P^{\,2}, \quad = \frac{3\pi}{4}(1.1RD)^2 = \frac{11.4}{4}R^2D^2$$

$$Vol = A\cdot t = \frac{11.4}{4}RD^2 \times \frac{P(1.1RD)}{S} = \frac{3.15PR^2D^3}{S}$$

$$Wt = \rho\cdot Vol = \frac{3.15\rho PR^2D^3}{S} \approx \frac{5.25\rho PR^3D^2}{S} \quad (\text{For } R \approx .6) \quad \text{(Ends)}$$

$$\text{Poppet-Assembly Wt} = \underline{\underline{\frac{16.6\rho R^3D^2}{S}}}$$

D. TOTAL VALUE WT $= \frac{\rho PD^3}{S}\left[4.56(1 - R^2) + 6.8R + 16.6R^3\right]$

Where: Wt = lb

ρ = Material, lb/in.^{3}

P = Proof Pressure, psi

D = Line ID, in.

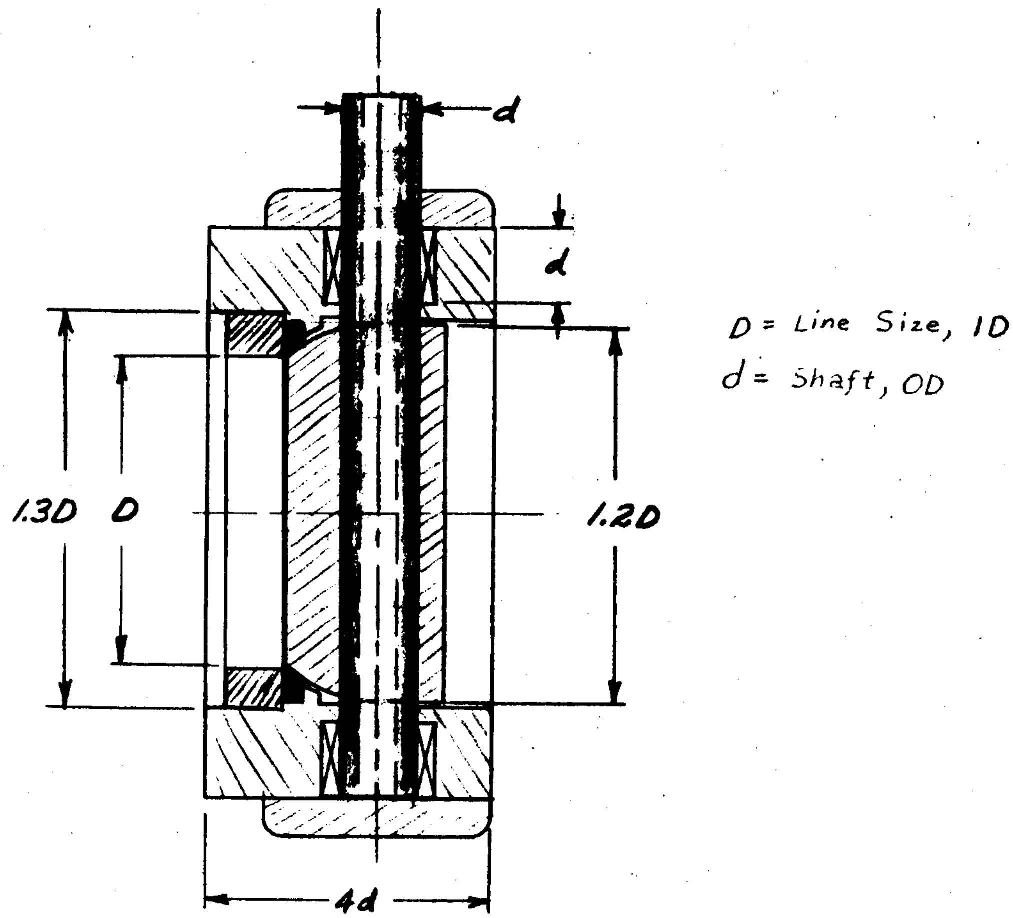
S = Material Stress at P

R = Throat/Line dia Ratio

$R = D_T/D$	$\left[4.56(1 - R^2) + 6.8R + 16.6R^3\right]$
0.8	14.51
0.6	10.02
0.5	8.57
0.4	7.34
0.3	6.48

IV. BUTTERFLY-VALVE WT VS SIZE AND PRESSURE

A. The weight of a typical butterfly valve is approximated in terms of line size, valve proof pressure, and material properties. The equations for weight will be derived for each part separately and summed to give the total valve weight. The high-pressure configuration, which was satisfactorily employed for Titan engines, will be assumed typical for the purpose of providing an analytical model. This typical configuration with proportional dimensions is shown on the following page.



B. ASSUMPTIONS AND CONDITIONS

1. The prime consideration for the selection of a butterfly valve for a high-pressure application is it's characteristically short plumbing-space (length) requirement and in-line configuration.

2. A straight-through one-piece shaft is desirable for purposes of fabrication and assembly.

3. For a high-pressure-application design, the shaft bearings are the weakest members of the valve; their successful functioning depends on maintaining a small allowable shaft deflection, which results in relatively low allowable stresses in the shaft and bearings. The bearings are assumed to be uniformly loaded, which is not actually the case because of the shaft deflection. The bearing-design stress is, therefore, set lower than the maximum bearing stress the material can withstand.

C. CALCULATIONS

1. Shaft Size and Weight

Shaft OD = d
Bearing ID = d
Bearing Length = d
Bearing Projected Area = d^2 per Bearing
Assume a Bearing Stress of 15 ksi*

Bearing Load = Pressure Force on the Blade

$$= P \times \frac{\pi}{4} (1.2D)^2 = 4.5\ PD^2$$

$$\text{Bearing Stress} = \frac{4.5PD^2}{4.2d^2} = 15 \text{ ksi}\ ,\quad \underline{\underline{d = .00614\ D\sqrt{P}}}$$

The length of shaft within the blade is best considered as part of the blade in the weight calculation. The length of shaft external to the blade is approximately 5 times the shaft diameter. If the shaft is hollow (ID = 1/2 OD) the volume and weight are:

$$\text{Vol} = 3/4\ \ \pi/4d^2 \times 5d = 3d^3 \text{ in.}^3$$

$$\text{Shaft Wt (External of Blade)} = 3\rho d^3 \quad \text{because } d = .00614\ D\sqrt{P}$$

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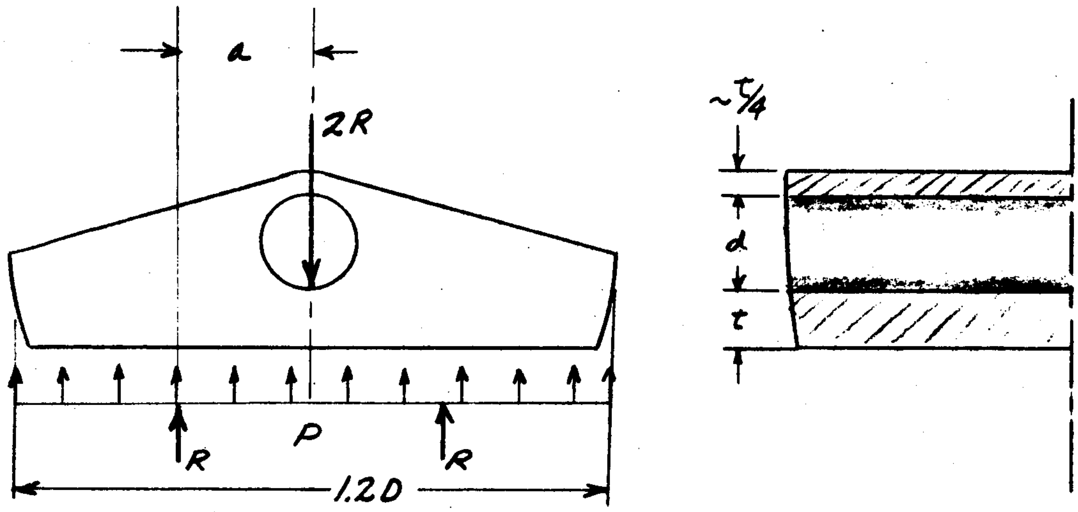
*Shaft deflection will reduce the effective bearing area 1/2 to 1/3 the total area, which results in maximum bearing stress of 30 to 40 ksi.

Shaft wt $= 6.9 \times 10^{-7}\ D^3\, P^{3/2}$

2. <u>Blade Size and Weight</u>

Assume: Blade OD = 1.2D

Blade - Stress Calculation:



Approximate Stress on Section (t wide)

The moment about the shaft center line is R x a, where:

$$R = \frac{\pi}{8} P\,(1.2D)^2 = .57PD^2$$ (Resultant Pressure Force on 1/2 of Blade)

$$a = \frac{2}{3}\ (1.2D) = .25D$$

Moment R.a = $.57PD^2 \times .25D = \underline{.14PD^3}$

This moment must be balanced by the moment (approximate):

$d\left[S(1.2dt)\right]$ Hence, $.14PD^3 = ds(1.2dt)$, where $d = .006\ D\sqrt{P}$

$$t = 3.5 \times 10^3\ \frac{D}{S}$$

Because the irregular shape of the blade would result in a lengthy formula for the volume and weight, the blade will be assumed to approximate a disc of diameter 1.2D and a thickness of t + 2/3d,

$$\text{Vol} \approx \frac{\pi}{4} (1.2D)^2 \times (t + 2/3d) = 1.1D^2 \left(3.5 \times 10^3 \frac{D}{S} + .004\, D\sqrt{P}\right)$$

$$\text{Vol} \approx 1.1D^3 \left(\frac{3.5 \times 10^3}{S} + .004\sqrt{P}\right)$$

$$\boxed{\text{Blade wt} = 1.1\rho D^3 \left(\frac{3.5 \times 10^3}{S} + .004\sqrt{P}\right)}$$

3. Body Size and Weight

From the sketch giving the assumed proportional dimensions, the average body ID is 1.25D, length = 4d. Assuming a bearing OD of 1.25d, the body wall thickness in terms of the allowable hoop stress is:

$$S = \frac{P \times 4d \times 1.25D}{2t \quad (4-1.25)d} = .91 \frac{PD}{t} \quad (t = \text{Body Wall Thickness})$$

$$t = \frac{.91\ PD}{S} \quad \text{Vol} = (1.5)^{*}\pi(1.25D + t)4dt, \text{ because } d = .00614\ D\sqrt{P}$$

$$t = \frac{.91\ PD}{S}$$

$$\text{Vol} = .105 \frac{P^{3/2}\ D^3}{S} \quad (1.25 + 0.91\ P/S)$$

$$\boxed{\text{Body wt} = .105\ \rho\ P^{3/2} \frac{D^3}{S} \quad \times \quad (1.25 + 0.91\ P/S)}$$

*Factor to account for bearing bosses and seal components (derived from Titan valve body).

4. Total Valve Weight

The total valve weight without an actuator is composed of the sum of the weights of the shaft, blade, and body, which is:

$$\text{Total Valve wt} = \left[6.9 \times 10^{-7} \rho D^3 P^{3/2}\right] \left[1.1 \rho D^3 \left(\frac{3.5 \times 10^3}{S} + .004\sqrt{P}\right)\right]$$

$$+ \left[.062 \rho P^{3/2} \frac{D^3}{S} (1.25 + 1.07\ P/S)\right]$$

$$\text{Total wt} = \rho D^3 \left[6.9 \times 10^{-7} P^{3/2} + \left(\frac{3900}{S} + .0044\sqrt{P}\right)\right.$$

$$\left. + .105 \frac{P^{3/2}}{S} (1.25 + P/S)\right]$$

where: ρ = Material Density, lb/in.^{3}

D = Line Size, ID

P = Proof Pressure, psi

S = Material Stress, psi (at P)

V. BALL VALVE WEIGHT VS SIZE AND PRESSURE AND MATERIAL

Reference Ball-valve* Dimensions: Port dia = 2-1/2 in.

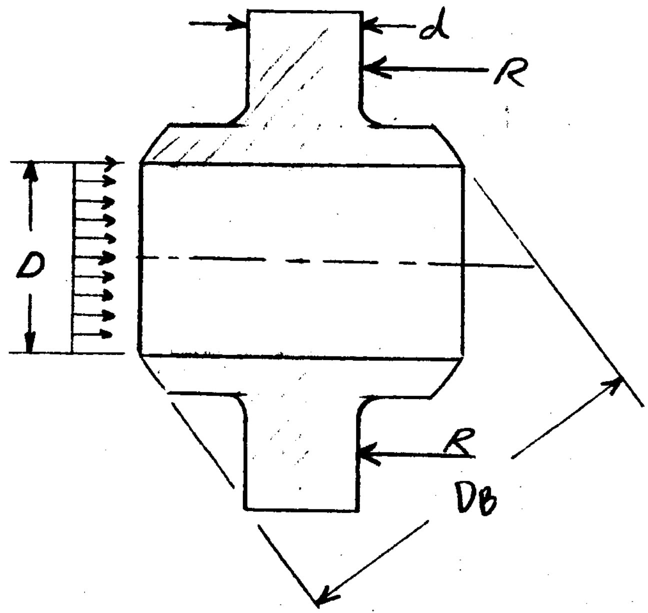
Ball dia = 5.25 in. Shaft dia = 3 in.

Proof Pressure = 5000 psi

To determine an equation for the weight of a ball valve in terms of port size and proof pressure, the relationship between principal dimensions will be determined parametrically as required by strength considerations. The coefficient to these relationships will be chosen to make them correlate to the reference valve described above, and to a zero pressure valve in which dimensions are specified by geometrical considerations only.

*Consolidated Controls Corp., PN 113W65A.

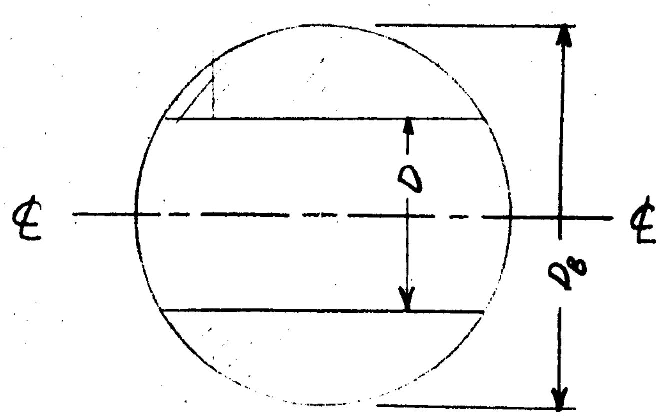
Ball and Shaft Configuration



Load on Bearings = $.785\ PD^2/2 = R = .393D^2 \cdot P$

approximate maximum bending moment (at ℄) $= R \cdot D_B/2$

The strength of the ball at the ℄ depends on D and D_B as follows:



Ball Sectional View

Section modulus $\approx .098\ D_B^{\ 3} - \frac{1}{6}\ D_B\ D^2$ (through ball ℄)

The exact section modulus can be determined by the following equation:

$$\frac{T}{c} = \frac{\left[\frac{A\ D_B^{\ 2}}{16}\left(1 + \frac{2\ \sin^3\theta\cos\theta}{\theta - \sin\theta\cos\theta}\right) - \frac{A\ D_B^{\ 6}\ \sin^6\theta}{144}\right]2}{D_{B/2}}$$

$$\cos\ \theta = \frac{D}{D_B} \qquad \theta = \text{ARC}\cos\frac{D}{D_B}$$

Bending stress at ℄

$$\frac{M℄}{Z℄} = \frac{.47\ D^2\ .\ P\left(D_{B/2}\right)}{.1\ D_B^{\ 3} - .16\ D_B\ D^2}\ \text{(approximate)}$$

$$S_B = \frac{M℄}{Z℄} = \frac{.47\ D^2 P \times D_{B/2}}{.1\ D_B^{\ 3} - .16\ D_B\ D^2} = \frac{.23\ D^2\ D_B}{.1\ D_B^{\ 3} - .16\ D_B\ D^2} =$$

$$= \frac{2.3\ P}{\frac{D_B^{\ 2}}{D^2} - 1.6} = S_B$$

$$D_B = D\sqrt{\frac{2.3P}{S_B} + 1.6}$$

If we take $S_B = 20{,}000$ psi $P = 5000$ psi

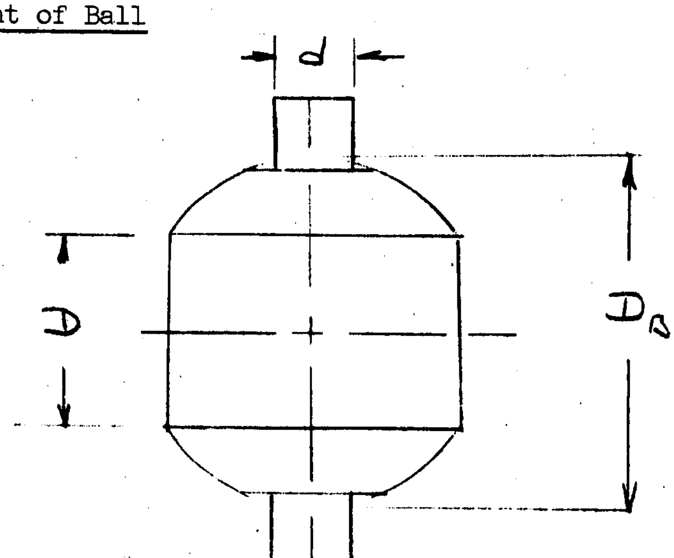
$$D_B = D\sqrt{\frac{2.3 \times 5000}{20{,}000} + 16} = D\sqrt{.575 + 1.6} = D\sqrt{2.175} = 1.47\ D$$

This is lower than the 1.7 value required for seals; therefore, the ball size in this case is determined by the space requirements for the seals. The bending stress, S_B, for the ball must be set sufficiently low that little deflection of the ball will take place.

Assume $D_B = 1.8\ D$, $P = 5000$ psi

$$S_B = \frac{2.3 \times 5000}{1.8^2 - 1.6} \quad \frac{11500}{3.24 - 1.6} = \frac{11500}{1.64} = 7000 \text{ psi} \quad \text{(conservative value)}$$

1. Weight of Ball



$$\text{Ball wt} = \frac{\pi}{6} D_B^3 - .85\ D_B \frac{\pi}{4} D^2 \rho$$

$$= \frac{\pi}{6} 1.8^3 D^3 - .85 \times 1.8\ D \frac{\pi}{4} D^2 \rho =$$

$$W_B = 3.04\ D^3 - 1.2\ D^3 \rho = \underline{\underline{1.84\ D^3 \rho_B}}$$

2. <u>Shaft Size, d</u>

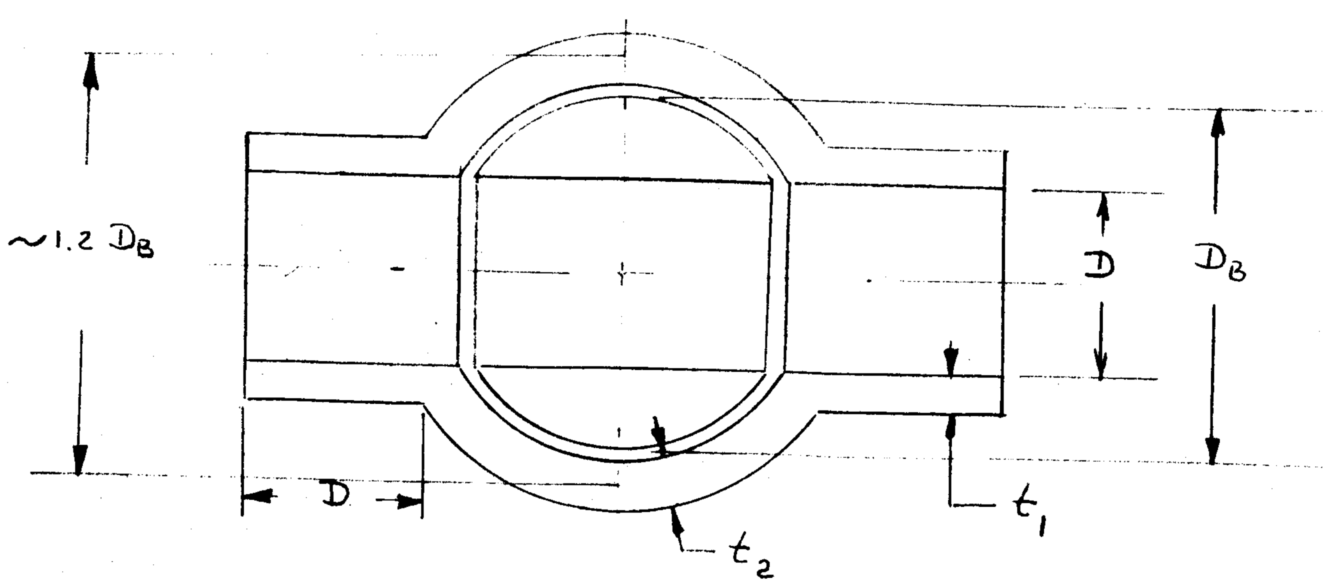
$R = .47\ PD^2$ (pressure-force load) S = 20,000-psi bearing stress

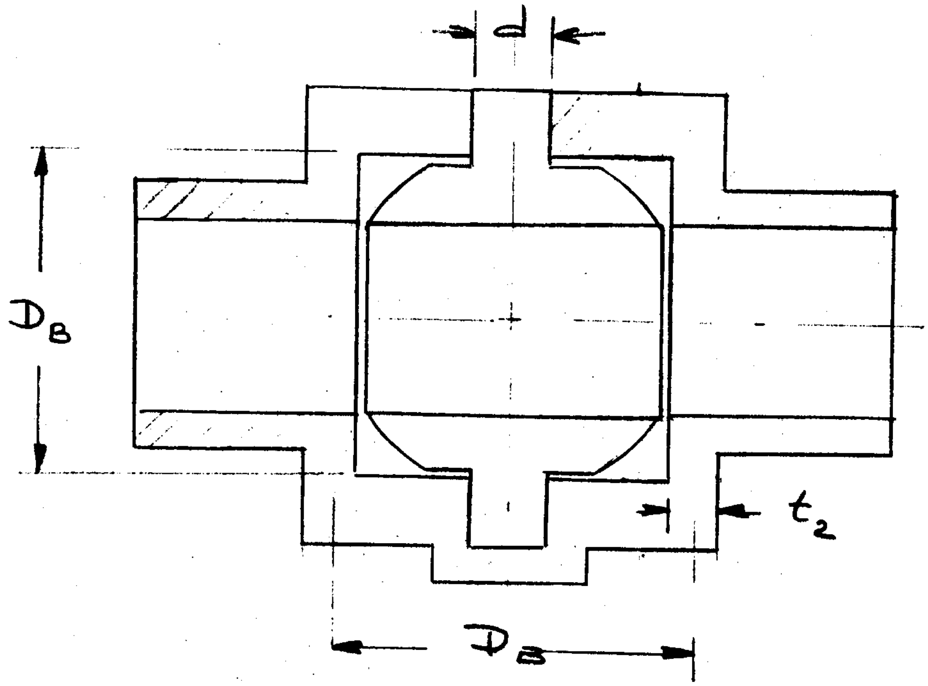
$$d = \sqrt{\frac{.47\ PD^2}{20{,}000}} = 4.35 \times 10^{-3} D \sqrt{P}$$ for bearings with length =

= dia = d

Shaft weight: $\frac{\pi}{4}\ d^2 \times d \times \rho = \frac{\pi}{4}\ d^3 \times \rho$

$$W_S = \underline{3.42 \times 10^{-3}\ \rho\ D \sqrt{P}}$$





Ball-Valve Housing Envelope

3. Body Weight (See ball-valve housing-envelope sketch.)

$$t_1 = \frac{PD}{2S} \qquad t_2 \quad \frac{P\,D_B}{2S} = \frac{1.8PD}{2S}$$

Volume: (approximately) = Vol (body sphere - openings + inlet cylinders)

$$1.2\pi D_B \,.\, t_2 \,.\, D_B - 2\,\frac{\pi}{4}\,D^2 \times t_2 +$$

$$2\pi(D + t_1)\,t_1 \times D$$

$$= 12.2\,D^2 t_2 - 1.57\,D^2 t_2 + 2\pi(D + t_1)\,t_1 D +$$

$$1.57\,(D_B + t_2)^2 d$$

$$= \frac{11PD^3}{S} - \frac{1.41\,PD^3}{S} + \frac{3.14\,PD^3}{S}\left(1 + \frac{P}{2S}\right)$$

$$\text{Vol} = \left(12.7 + \frac{1.57P}{S}\right)\frac{PD^3}{S}\ \text{in.}^3$$

Weight of body

$$W_B = \left(12.7 + 1.57\,\frac{P}{S}\right)\frac{\rho_B PD^3}{S}$$

4. Total Weight of Valve = $W_B + W_S + W_B$

$$W_V = 1.84\,\rho_B D^3 + 3.42 \times 10^{-3}\rho_S'\,D\sqrt{P}$$

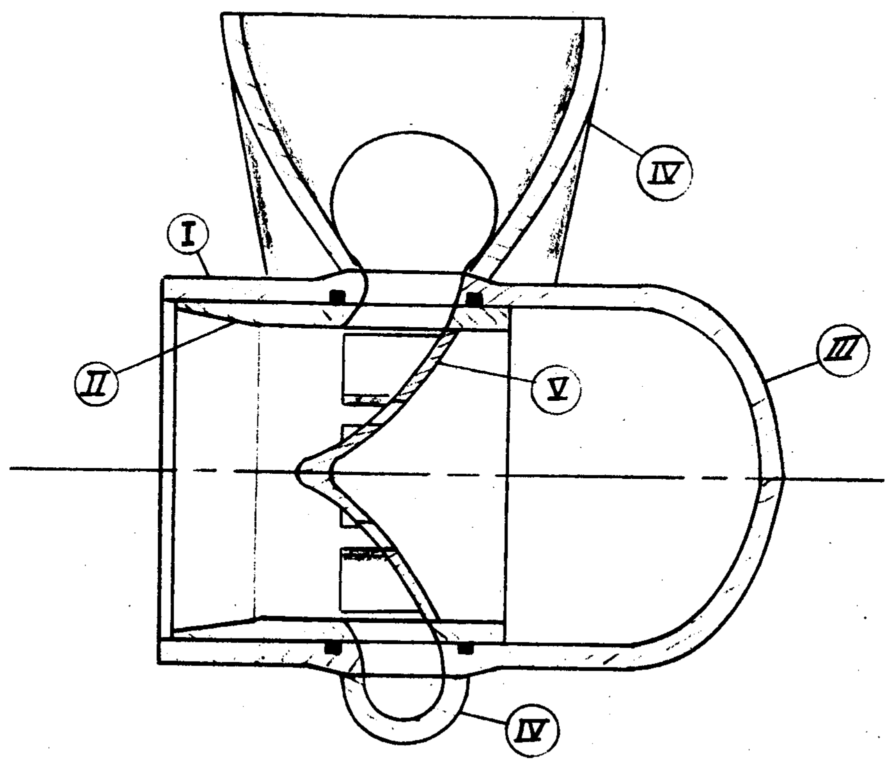
$$+ \left(12.7 + 1.57\,\frac{P}{S}\right)\rho_B\,\frac{PD^3}{S}$$

Neglecting the second term and simplifying the last term

$$\text{Approx } W_V = 1.84\,\rho\,D^3\left(1 + \frac{7P}{S}\right)\text{lb}$$

VI. ANGLE SLEEVE VALVE--WEIGHT AS A FUNCTION OF SIZE, PRESSURE, AND MATERIAL

Assuming the following model for weight analysis, the weight of the various valve parts will be determined individually according to stress considerations, and combined to give the total valve weight.



Part I

$d_{ave} = 1.1D$, length $= l = 2.5D$

t = thickness $= \frac{PD}{2S}$, (assume reinforcement in part area cancels the volume of the metal removed to make the part)

Volume of material $= \pi\, dlt = \pi \times 1.1D \times 2.5D \times \frac{PD}{2S} = \frac{4.4\, PD^3}{S}$

Weight = volume x density $= \frac{4.4\, PD^3 \rho}{S}$

Part II

$$d_{ave} = .9D, \quad l = 2D, \quad t = \frac{P(.9D)}{2\ S}$$

$$\text{Volume} = \pi d\ l\ t = .9\,\pi D \times 2D \times \frac{.9PD}{2\ S} = \frac{2.6\ PD^3}{S}$$

$$\text{Weight} = \frac{2.6PD^3\rho}{S}$$

Part III

$$\text{Area} = \pi D^2, \quad t = t\ (\text{body} = \frac{PD}{2S}$$

$$\text{Volume} = 1.6\ \frac{PD^3}{S}, \quad \text{Weight} = \frac{1.6PD^3\rho}{S}$$

Part IV

$$\text{Average torus flow area} = .785D^2 \times \sim 3/8 = .29D^2$$

$$\text{Approximate surface area} = 1.2D\ (7/8) \times \pi(1.75D) = 5.8D^2$$

$$\text{Approximate average thickness} = \frac{1.75\ PD}{2S}\ (1.5)^* = \frac{1.3PD}{S}$$

$$\text{Volume} = \frac{7.6PD^3}{S}, \quad \text{Weight} = \frac{7.6PD^3\rho}{S} \quad \text{(torus collector)}$$

Transition section to round exit section:

$$D = D, \quad l = .5D\ \text{(approximate)} \quad \text{average } t = \frac{PD}{2S}\ (1.25) = \frac{.63PD}{S}$$

$$\text{Volume} = \frac{1 \times PD^3}{S} = \frac{.79PD^3}{S}, \quad \text{Weight} = \frac{1 \times PD^3\rho}{S}$$

*Factor to allow for the stress concentration at point of attachment.

Part V

Surface Area $= \pi D^2$ (Approximate), $t = \left(\frac{1}{2}\right) \times \frac{PD}{2S} = \frac{PD}{4S}$

Volume $= \frac{.8\ PD^3}{S}$ = Weight $= \frac{.8PD^3 \rho}{S}$

Total Valve Weight:

Sum of part weights $= (4.4 + 2.6 + 1.6 + 7.6 + 1.0 + .8)\ \frac{PD^3 \rho}{S}$

Total Valve Weight $= \frac{18\ PD^3 \rho}{S}$

APPENDIX D

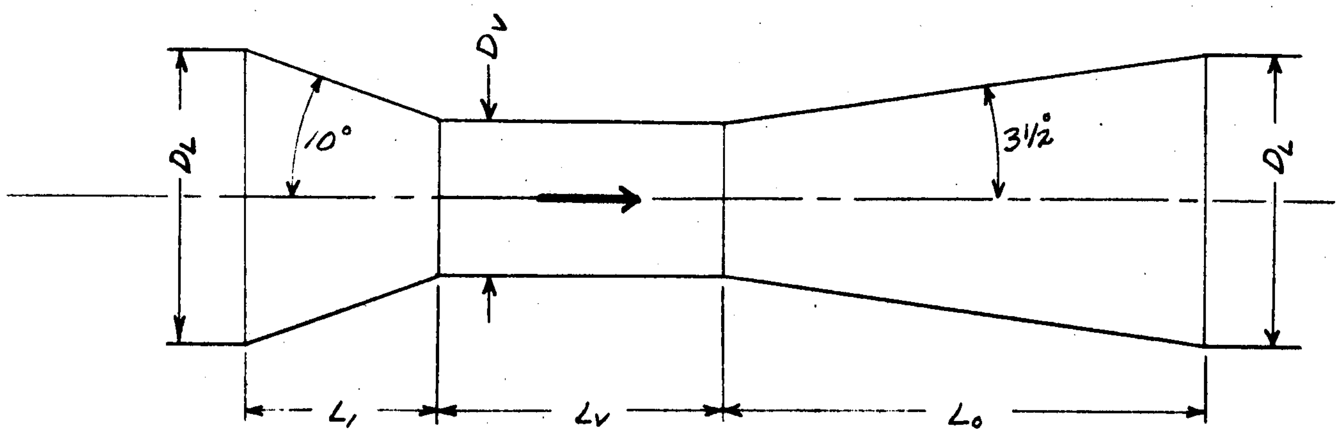
DESIGN OF VENTURI SECTIONS FOR INDIVIDUAL VALVES

I. PRESSURE DROP

The purpose of applying venturi inlet and outlet sections to valves is to permit the use of a valve that is considerably smaller than line size. The general benefits incurred in using a smaller valve are: Smaller static-pressure forces; hence, less deflection, bearing load, etc.; less cost for the valve; and probably less valve weight if venturi sections are considered part of the line.

Disadvantages associated with using this system are: longer required in-line length and greater pressure loss due to increased flow velocity through the valve.

General layout



Note: The inlet and outlet cone angles are maximum for minimum loss: Vennard-*Fluid Mechanics*, p. 297.

Let K_V = Valve kinetic loss factor, i.e.,

$$\Delta P_V = K_V \cdot \frac{\rho V^2}{2g\ 144} = K_V \cdot \frac{3.64\dot{w}^2}{\rho D_V^4}$$

From the above equation it may be seen that the pressure loss through a valve is inversely proportional to the diameter to the 4th power. Then, if $D_V = 1/2 D_L$, the pressure loss will be $2^4 = 16$ times the pressure loss of a line-size similar valve.

It is, therefore, required that a low-loss-type valve be employed in this application, such as a ball, in-line sleeve, or poppet-type valve (where $K_V < .5$).

The pressure loss due to changing the velocity through the venturi sections is $\Delta P_{V'} = K_{V'} \cdot \frac{3.64\dot{w}^2}{\rho D_V^4}$, where $K_{V'}$ is the kinetic loss factor for the venturi sections.

For an exit (diffuser) section with a 7° included angle, the value of $K_{V'} = 0.08\ (1/D_V^4 - 1/D_L^4)\ D_V^4 = .08 \left[1 - \left(\frac{D_V}{D_L}\right)^4\right]$

For an inlet section with a 20° included angle, the pressure loss is:

$$K_{V''} \frac{3.64\dot{w}^2}{\rho D_V^4}$$

Where $K_{V''} = .02$*

Total loss (inlet, valve, outlet):

$$\sum \Delta P = (K_{V''} + K_V + K_{V'}) \frac{3.64\dot{w}^2}{\rho D_V^4}, \text{ where: } K_{V''} = .02$$

K_V = Valve loss factor

$K_{V'} = .08 \left[1 - \left(\frac{D_V}{D_L}\right)^4\right]$

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* "Vennard" Fluid Mechanics, p. 215.

Total pressure loss

$$\sum \Delta P_L = (K_{V''} + K_V + K_{V'}) \frac{3.64\dot{w}^2}{\rho D_V^4} = \left\{ .02 + K_V + .08 \left[1 - \left(\frac{D_V}{D_L} \right)^4 \right] \right\} \cdot \frac{3.64\dot{w}^2}{\rho D_V^4}$$

Let $\frac{D_V}{D_L} = R$, $\left(\frac{D_V}{D_L} \right)^4 = R^4$

$$\sum \Delta P = \left[.02 + K_V + .08 \left(1 - R^4 \right) \right] \cdot \frac{3.64\dot{w}^2}{\rho D^4},$$

$$\boxed{\text{Let } \sum \Delta P = K_L \cdot \frac{3.64\dot{w}^2}{\rho D^4}} \quad K_L = .02 + .08 \left(1 - R^4 \right) + K_V$$

Where: $\sum \Delta P$ = Pressure loss of inlet, valve, and diffuser combined

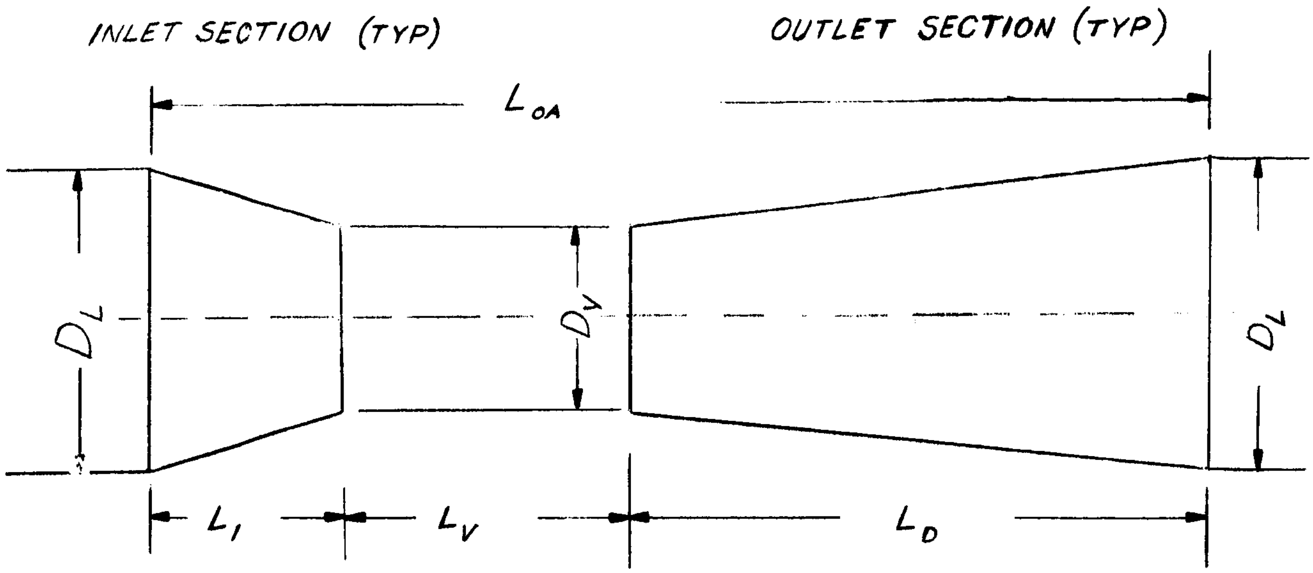
R = D_V/D_L = < 1

K_V = Valve Kinetic Energy Loss Coefficients

$\dot{w}$ = Fluid flow, lb/sec

ρ = Fluid density, lb/ft^3

D_L = Line size, ID, in.



Notation: D_L = Line size, ID

D_V = Valve inlet and outlet, ID

L_I = Venturi-inlet section length

L_V = Valve length

L_D = Venturi-diffuser section length

Assume: Inlet section included angle = 20°, Diffuser section included angle = 7° — required for minimum loss

$R = D_V/D_L$ = Diameter ratio

Expressing L_I and L_D in terms of D_L, ($\theta_I = 20°$, $\theta_D = 7°$) and R

$$L_I = \frac{\frac{1}{2}D_L(1-R)}{\tan \frac{1}{2}\theta_I} = 2.84\, D_L\,(1-R)\;,\quad L_D = \frac{\frac{1}{2}D_L(1-R)}{\tan \frac{1}{2}\theta_D} = 8.2 D_L(1-R)$$

Overall length = $L_I + L_V + L_D$

$$\boxed{L_{OA} = 11.04\, D_L\,(1-R) + L_V} = \text{Overall length}$$

Assume that the wall thickness of the venturi sections is constant from inlet to outlet. Although hoop-stress considerations would allow a thinner wall at the throat, bending stress and axial loads will tend to off set the decrease in hoop stress in this region.

Required wall thickness, based on hoop stress at maximum diameter:

$$t = \frac{P \cdot D_L}{2S}$$

, t = Wall thickness, in.

D_L = Line ID, in.

P = Proof pressure, psi

S = Design stress at P, psi (hoop stress)

Assuming mean diameter $\equiv$ ID + t $\approx$ ID x 1.04

$$Vol = \frac{1.04\pi t}{2} (D_L + D_V)(L_I + L_D)$$

Letting $R = D_V/D_L$ = dia ratio, where $DV = RD_L$

$$Vol = 1.63 \left(\frac{PD_L}{2S}\right) D_L (1+R) (L_I+L_D) = .815 \frac{PD_L^2}{S}(1+R) \overbrace{(11.04D_L)(1-R)}^{L_1+L_D}$$

$$\underline{Vol = \frac{9PD_L^3}{S} (1-R^2)} \text{ in.}^3$$

(if D_L is in in., P and S are in psi)

$WT = \rho_m \times Vol$ where ρ_m = Material Density, lb/in.^{3}

$$\boxed{WT = \frac{PDL^3}{S} (I-R^2)}$$

WT = Inlet and Diffuser - Section Wt Not including Valve Wt.

ρ_m = Material Density, lb/in.^{3}

P = Proof Pressure, psi

D_L = Line ID

S = Material Hoop Stress at P, psi

R = Ratio of Valve Dia/Line ID

APPENDIX E

SIZE CALCULATION FOR THE AJ-1
ENGINE THRUST VECTOR CONTROL SYSTEM

I. THRUST-VECTOR-CONTROL REQUIREMENT

The maximum desired lateral (side) thrust required of the AJ-1 thrust-vector-control (TVC) system for heavy steering is 10%* of the axial thrust. With proper design and placement of the injectant nozzle, a lateral amplification of 2.5 may be assumed. That is:

$$\frac{\text{Lateral thrust}}{\text{Axial thrust}} = 2.5 \times \frac{\text{Lateral flowrate}}{\text{Axial flowrate}}$$

The required TVC flowrate to produce 10% side thrust is

$$\dot{w}_{TVC} = \dot{w}_{axial} \times 0.1 \times \frac{1}{2.5} = 0.04\ \dot{w}_{axial}$$

For this engine, $\dot{w}_{axial}$ = 15,650 lb/sec

$$\dot{w}_{TVC} = 0.04 \times 15{,}650 = 628 \text{ lb/sec}$$

II. TVC SYSTEM

The proposed system functions by bleeding hot gas at approximately 1900°R from the primary combustor to four control valves (one for each quadrant). From the control valves, the gas flows to distribution manifolds that supply five injectant nozzles per quadrant. The following analysis is to determine the size of the control valves and injectant nozzles necessary to meet the system requirements.

III. ASSUMPTIONS USED FOR SIZE CALCULATIONS

The total engine propellant flow rate is 15,650 lb/sec.

$$\dot{w}_f = 2240 \text{ lb/sec}, \quad \dot{w}_o = 13{,}410 \text{ lb/sec}$$

---

* Assumed value.

The TVC maximum flow rate is 4% of the total flow rate: $0.04 \times 15{,}640 = 628$ lb/sec.

Mixture ratio of TVC gas = 0.915

The flow rate for the TVC system is as follows:

$$\dot{w}_f = \frac{628}{1.915} = 328 \text{ lb/sec}, \quad \dot{w}_o = 628 - 328 = 300 \text{ lb/sec}$$

A. GAS COMPOSITION

Reaction: $H_2 + 1/2\ O_2 \rightarrow H_2O$

By molecular weights: $2H_2 + 1/2(32)O_2 = 18H_2O$ or $1H_2 + 8O_2 = 9H_2O$

Because the actual mixture ratio is very fuel rich, only part of the fuel will be burned. The combustion products are a mixture of water vapor and GH_2 in the following proportion:

Weight of H_2 reacted = 1/8 (weight O_2) = 1/8(300 lb/sec) = 37.5 lb/sec

Weight of unburned H_2 = 328 - 37.5 = 290.5 lb/sec
Weight of product (H_2O) = 300 + 37.5 = 337.5 lb/sec

B. DENSITIES

$\rho = \frac{P}{RT}$, $R_{H_2} = 773$,

$P = 3570$ psi $= 5.15 \times 10^5$ lb/ft^2, $T = 1850°R$*

$R_{H_2O} = 85.8$,

* Assumed conditions at exit from primary combustor.

$$\rho_{H_2} = \frac{5.15 \times 10^5}{773 \times 1850} = 0.36 \text{ lb/ft}^3$$

$$\rho_{H_2O} = \frac{5.15 \times 10^5}{85.8 \times 1850} = 3.24 \text{ lb/ft}^3$$

C. VOLUME FLOW RATES

$$\dot{V}_{GH_2} = \frac{\dot{W}f}{\rho f} = \frac{290.5 \text{ lb/sec}}{0.36 \text{ lb/sec}^3} = 805 \text{ ft}^3\text{/sec } GH_2$$

$$\dot{V}_{H_2O} = \frac{\dot{W}_{H_2O}}{\rho_{H_2O}} = \frac{337.5 \text{ lb/sec}}{324 \text{ lb/ft}^2} = 104 \text{ ft}^3\text{/sec}$$

D. OTHER CONDITIONS

Total Volume Flow Rate $= \Sigma \dot{V} = 805 + 104 = 909 \text{ ft}^3\text{/sec}$

Density of gas $= \frac{\dot{W}}{\dot{V}} = \frac{628 \text{ lb/sec}}{909 \text{ ft}^3\text{/sec}} = 0.690 \text{ lb/ft}^3$ (static supply)

Specific-heat ratio $= C_p/C_v = 1.35$*

Critical-pressure ratio $= \left(\frac{2}{K\ 1}\right)^{\frac{K}{K-1}} = (0.85)^{3.86} = 0.534$

Sonic Velocity (supply conditions) $= KgP/\rho$

$P = 3570 \times 144 = 5.15 \times 10^5 \text{ lb/ft}^2$, $\rho = 0.69 \text{ lb/ft}^3$

$$R_{comb} = \frac{P}{\rho_c T} = \frac{5.15 \times 10^5 \text{ PSF}}{0.69 \times 1850°R} = 400$$

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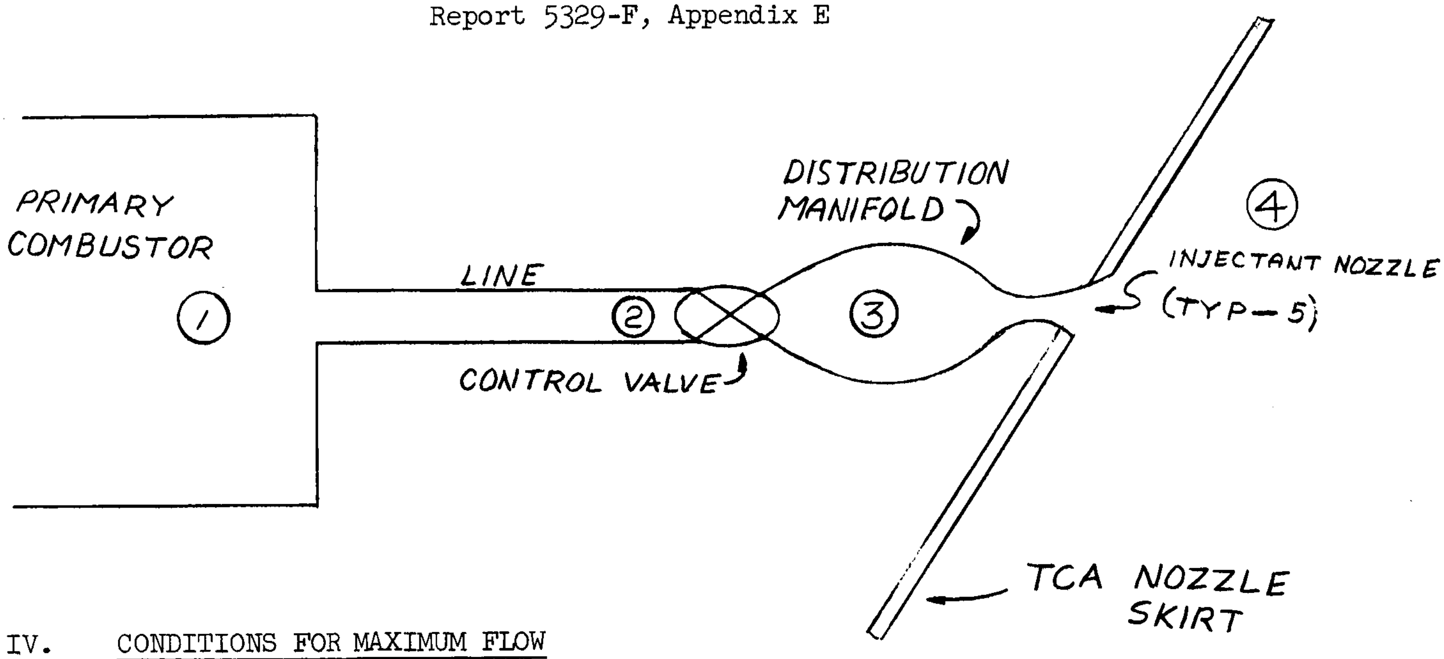
* Data from engine analysis.



IV. CONDITIONS FOR MAXIMUM FLOW

Condition 1 (Static Conditions in Primary Combustor)

P_1 = 3570 psia (static pressure)

T_1 = 1900°R

Condition 2 (At Control-Valve Inlet)

P_2 = 3500 psia (total, assuming 70-psi line loss)

T_2 = 1850°R (total temperature assuming 50° drop due to line heat loss)

Condition 3 (At Injectant-Nozzle Manifold)

For sonic flow through the control valve:

$P_3/P_2 = 0.534$, $P_3 = 0.534\ P_2 = 1870$ psi (static pressure)

$$T_3 = T_1\ (P_3/P_1)^{\frac{K-1}{K}} = 1900\ (1870/3470)^{0.26} = 1610°R$$

Condition 4 (Downstream from Injectant Nozzles)

$P_4 = P_c$ at injectant nozzle = 50 psia

T_4 (injectant stream) $= T_3 \, (P_4/P_3)^{\frac{K-1}{K}} = 1610(50/1870)^{0.26} = 630°R$

V. SIZE OF SONIC INJECTANT-NOZZLE THROAT

$$A_T = \frac{\dot{w}\sqrt{T}}{P \sqrt{\frac{gK}{R}\left(\frac{2}{K-1}\right)\frac{K\ 1}{K-1}}} \text{ exp}$$

Where:

$\dot{w}$ = lb/sec

T = upstream temperature, °R

P = upstream pressure, lb/ft^2(absolute)

g = 32.2 ft/sec^2

K = 1.35

R = 400

$$A_T = \frac{5.25\ \dot{w}\ \ T}{P}$$

Maximum flow conditions:

$\dot{w}$ = 628 lb/sec for 5 nozzles = 126 lb/sec/nozzle

T_3 = 1610°R

P_3 = 1870 psia = 2.7 x 10^5 lb/ft^2

$$A_T = \frac{5.23 \times 126 \times \sqrt{1610}}{2.7 \times 10^5}\ 0.0978 \text{ ft}^2 = 14.1 \text{ in.}^2$$

Throat diameter = 4.25 in.

VI. MINIMUM FLOW CONDITION (MACH 1 THROAT)

$$P_3 = P_4/0.534 = 50/0.534 = 94 \text{ psia} = 1.35 \times 10^4 \text{ lb/ft}^2$$

$$T_3 = T_2 \left(P_3/P_2\right)^{\frac{K-1}{K}} = 1850\ (94/3500)^{0.26} = 720°R$$

$$A = 0.098 \text{ ft}^2 \text{ (calculated above)}$$

$$\dot{w}_{min} = \frac{AP}{5.23\ T} = \frac{0.098 \times 1.35 \times 10^4}{5.23 \quad 720} = 9.45 \text{ lb/sec/nozzle}$$

$$\times 5 = 71.5 \text{ lb/sec}$$

Maximum/Minimum sonic flow ratio = 126/9.45 = 13.3:1*

VII. SIZE OF TVC MODULATING VALVE

$$\dot{w}_{max} = 628 \text{ lb/sec}$$

Upstream conditions:

Total Pressure = 3500 psia = P_2

Total Temperature = 1850°R = T_2

Downstream conditions:

Static Pressure = 1870 psia = P_3

Static Temperature = 1610°R = T_3

Required Flow Area:

$$A = \frac{\dot{w}\sqrt{T_2}}{P_2\sqrt{\frac{gK}{R}\left(\frac{2}{K+1}\right)^{\frac{K+1}{K-1}}}},$$

Where:
$\dot{w}$ = 628 lb/sec
T_2 = 1850
P_2 = 3500 psia = 5.04×10^4 lb/ft^2
K = 1.35
R = 400 (for mixture)

* Not to be confused with max/min side-thrust ratio.

$$A = \frac{628 \sqrt{1850}}{5.04 \times 10^5 \sqrt{\frac{32.2 \times 1.35}{} \left(\frac{2}{2.35}\right)^{6.7}}} = 0.282 \text{ ft}^2 = 40.5 \text{ in.}^2 \text{ flow area}$$

Required diameter (equivalent covergent-nozzle diameter):

$$= \frac{40.5}{0.785} = 7.2 \text{ in. dia *}$$

VIII. NOZZLE FOR MACH 2 INJECTION

Upstream conditions:

$P = P_3 = 1870$ psia

$T = T_3 = 1610°R$

Downstream condition:

$P = P_4 = 50$ psia

$T_4 = T_3\ (P_4/P_3)^{0.26} = 700°R$ (injectant)

$V_{sonic-4} = \sqrt{1.35 \times 32.2 \times 400 \times 700} = 3500$ ft/sec, Mach 2, V=7000 ft/sec

$$\rho_4 = \frac{P}{RT} = \frac{50 \times 144}{400 \times 700} = 0.027 \text{ lb/ft}^3$$

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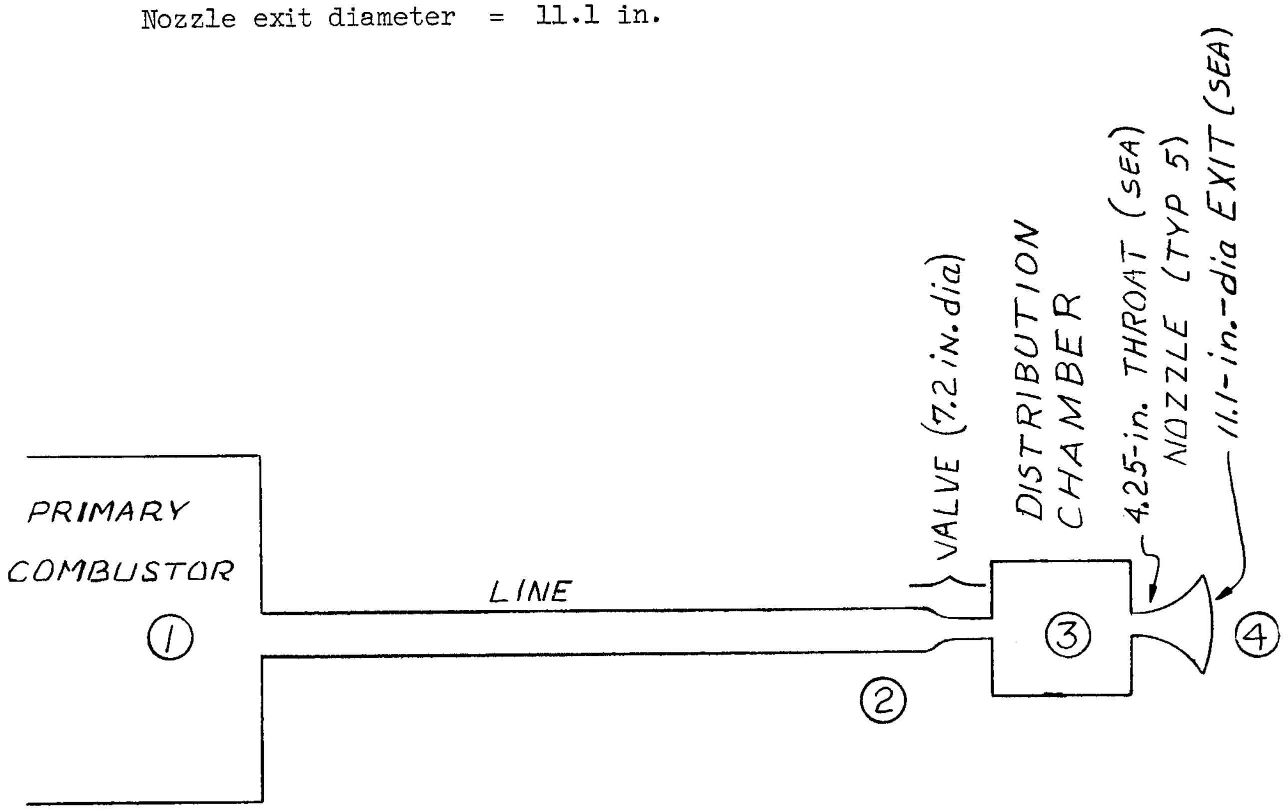
* Assuming $C_D = 1$.

$$A_e = \frac{\dot{w}}{\rho_e V_e} = \frac{126}{0.027 \times 7000} = 0.67 \text{ ft}^2 = 96.5 \text{ in.}^2$$

Nozzle exit diameter = 11.1 in.

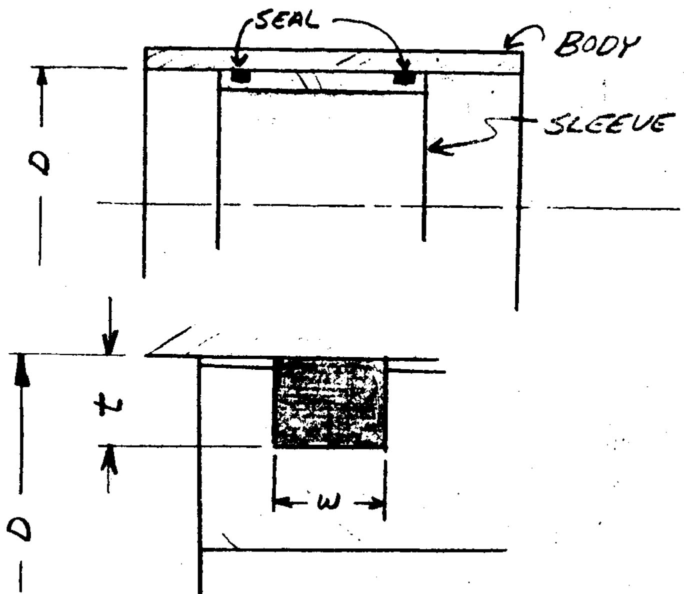


APPENDIX F

SEAL AND BEARING DRAG DETERMINATIONS

I. IN LINE AND ANGLE SLEEVE VALVE

The seal configuration of these valves is as follows:



SEAL DETAIL

In conventional seals (i.e., omni, bal, O-ring, lip seals) the seal is forced into contact with the sliding surface by the pressure acting on the area w x π D. The seal drag is the product of this radial force times the seal-to-bore coefficient of friction. The friction force is therefore a function of the following variables:

1. ΔP across the seal
2. seal (bore) dia = D
3. seal width = w
4. coefficient of friction = μ
5. seal installed load (may be defined as drag at zero pressure)

An expression for seal drag force in terms of size, D, and pressure, ΔP, may be derived by assuming:

Let w = function of D $\approx .04D$ for 4 in. $< D <$ 36 in. (approximate range)

Assume μ = .04 (approximate μ of Teflon)

$$\text{Drag Force} = F_D = P \times W \pi D \times \mu = \underline{.00503\ PD^2}$$

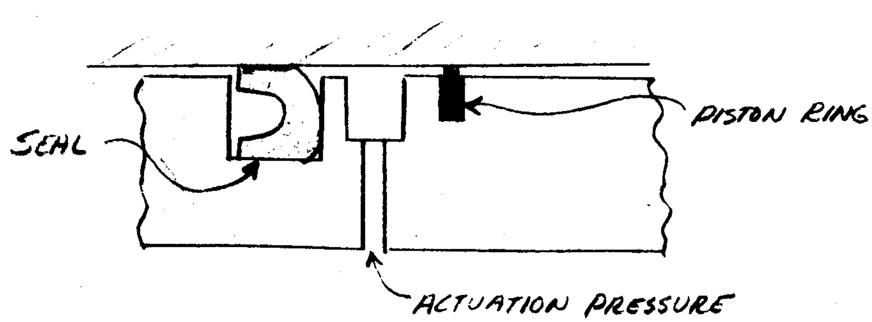
where: F_D = axial drag force, lb, due to one seal
P = ΔP across seal, psi
D = seal dia $\approx$ line size (ID) in.

Because the sleeve valves have 2 seals,

$$\boxed{F_D = F_1 + .011\ PD^2} \quad \text{for 2 Teflon seals, width} = .04D$$

where: F_D = axial seal drag, lb
P = seal ΔP, psi
D = seal $\approx$ line dia, in.
F_1 = installed seal drag, no ΔP

It may be noted that the drag force may be reduced by decreasing the seal ΔP during actuation. This requires porting pressure to the low-pressure side of the seals just prior to actuation, i.e.,



II. BUTTERFLY VALVE

The required actuator torque for a butterfly valve is a function of the following three variables:

1. seal friction
2. bearing friction
3. dynamic flow force on the butterfly blade

Assuming:

blade dia $= D_B = 1.2\ D$
seal width $= .04\ D = w$
seal coefficient of friction $= \mu = .04$ (Teflon)
bearing load $= 20$ ksi
bearing area $= 2d^2$, d = shaft dia
bearing coefficient of friction $= .04$ (Teflon composite)

SEAL FRICTION

Area "seeing" pressure is: $A = \pi D_B w = .04\pi \cdot 1.2\ D^2 = .15\ D^2$

Friction torque due to 1 seal $= PA \cdot 1/2\ D_B \cdot \mu = \underline{.0036\ PD^3 \text{ in.-lb}}$

Friction torque due to bearings:

For bearing stress = 20 ksi, $20 \text{ ksi} \times 2\ d^2 = \pi/4\ PD_B^2$

$$d = \sqrt{2.8 \times 10^{-5}\ PD^2} = .0053\ D\sqrt{P}$$

Bearing force $= \pi/4\ P\ (1.2\ D)^2 = 1.13\ PD^2$

Bearing torque $= \text{force} \times d/2 \times \mu = 1.13\ PD^2 \times .0026\ D\sqrt{P} \times \mu$

$= \underline{12 \times 10^{-5}\ P^{3/2}\ D^3 \text{ in.-lb}}$

Torque due to dynamic flow forces:

Ref: "Control Valves by C. S. Beard", p. 39, 40

Torque $= T = G \Delta P D_B^3$, because $D_B = 1.2\ D$, $T = 1.73\ G \Delta P D^3$

From p. 39 of the Ref, T_{max} for a 10-in. valve at 1 psi $=$ 320 in.-lb

hence, $G = \dfrac{T}{\Delta P D^3} = \dfrac{320}{1000} = 0.32$

Although this value of G is for a particular shape of butterfly blade, and varies with blade design, it may be assumed that high-pressure valves (having thicker blades) will not have higher torque values than this value assumed to be designed for less than 1000 psi (and, therefore, has a relatively thin blade). Using this value of G for an approximation of dynamic flow torque will undoubtedly yield torque values higher than would be attained in a high-pressure, thick-bladed valve specifically designed to minimize this torque. Therefore,

$$T \leq .32 \times \Delta P D^3 \text{ in.-lb} = \underline{.32\ D^3 \Delta P} \quad \text{(due to flow forces)}$$

Because the torque is proportional to the valve ΔP, it will depend on the system flow characteristics and rate of opening; this maximum torque occurs when the valve is 80% open and is zero when the valve is full open or full closed.

The total torque required (for the actuator), therefore, varies with valve position and reaches a maximum at 80% open (if the dynamic flow torque exceeds static seal plus bearing torque, as it usually does).

The required actuator torque at various stages of opening are:

0% open (break-away torque) = seal + bearing torque =
$.0036\ PD^3 + 12 \times 10^{-5}\ P^{3/2}\ D^3 = \underline{.0036\ PD^3\ (1 + .03\ \sqrt{P})}$

80% open, total torque = bearing + dynamic flow torque (on opening) =
$12 \times 10^{-5}\ \Delta P^{3/2}\ D^3 + .32\ D^3\ \Delta P$,

assuming that the downstream volume is sufficiently large that the valve ΔP is equal to the upstream pressure during opening (i.e., zero downstream pressure during opening) $\Delta P \approx P$, therefore torque at 80% open $\approx 12 \times 10^{-5}\ P^{3/2}\ D^3 + .32\ PD^3$.

$\underline{\text{Torque} \approx .32\ PD^3}$ (approximation) if $\Delta P \approx P_{upstream}$

It must be noted that if the downstream pressure is significant during valve opening, the dynamic and bearing torque will be less than the value given in the above formula; the torque must be calculated using the equation:

$\underline{\text{Torque} \approx .32\ D^3\ \Delta P}$ (max at 80% open)

<u>Sample calculation</u> Assume D = 10 in. System pressure P = 1000 psi
Initial breakaway torque = $.0036 \times 1000 \times 10^3\ (1 + .03\ \sqrt{1000})$ = 7000 in.-lb
Bearing torque alone = $12 \times 10^{-5} \times 1000^{3/2} \times 10^3$ = 3800 in.-lb.

Dynamic torque = $.32 \times 10^3 \times \Delta P$ = $320\,\Delta P$ in.-lb = $26.7\,\Delta P$ ft-lb

For values of this size and ΔP, the blade should be shaped to reduce this torque, which tends to close the valve.

III. BALL VALVE

Assume: Seal dia $= D_S \approx D$ (can be closely approximated)

Ball dia $= D_B = \sqrt{\frac{20P}{S_B} + 1.6}$

Shaft dia $= d$

Bearing area $= 2d^2$

Bearing and seal coefficient of friction $= .04$ (Teflon)

Bearing stress $= 20$ ksi

Shaft size:

$P \times \pi D_S^2 = 2d^2 \times 20$ ksi, $d = 4.43 \times 10^{-3} D\sqrt{P}$ (for 20 ksi brg. stress)

BEARING FRICTION TORQUE

Bearing load $= \pi/4\ D^2 \cdot P = .785\ PD^2$

Bearing torque $=$ load x radius x $\mu = .785\ PD^2 \times 2.21 \times 10^{-3} D\sqrt{P} \times .04$

Torque $= \underline{7 \times 10^{-5} P^{3/2} D^3 \text{ in.-lb}}$ (due to bearing friction)

SEAL FRICTION TORQUE

Seal area that "sees" pressure $= \pi D \times w$, $D =$ line size, $w =$ seal width. Assume seal width $= .04$ D, Area $= A_S = .04\pi D^2 = .126\ D^2$ in.^{2} Pressure force on seal $= P \cdot A_S = .126\ PD^2$

Seal friction torque $=$ force x 1/2 D x $\mu = .126\ PD^2 \times D/2 \times .04$ in.-lb

Torque $= \underline{2.5 \times 10^{-3} PD^3 \text{ in.-lb}}$ (due to seal friction)

Total friction torque $= \underline{\underline{2.5 \times 10^{-3}\, PD^3 \left(1 + .03 \sqrt{P}\right) \text{ in.-lb}}}$ due to seal and bearing friction (Teflon bearings, and seals $\mu = .04$).

where: P = static ΔP, psi

D = line = ball port dia, in.

APPENDIX G

PARAMETRIC STUDY OF LINE MATERIAL PROPERTIES

A simplified analysis was performed to establish the effect of allowable stress and elastic modulus on the flexibility of a continuous (unjointed) line. The analyzed line takes the form of a simple, horizontal cantilever beam loaded at the end by a force, F, producing a vertical deflection, Δ . In the initial analysis, the effect of pressure is considered only as the factor that determines wall thickness for a given, allowable hoop stress.

$$t = \frac{Pr}{S}$$

where: t = wall thickness, in.

P = internal pressure, psi

r = mean radius, in.

S = allowable hoop stress, psi

Other pressure effects, such as the axial force resulting from the pressure, are neglected. It is assumed that the structure retains a circular cross section when deflected. Flexibility (f), in this instance, is defined as the deflection per unit length,

$$f = \frac{\Delta}{L}, \text{ dimensionless} \qquad \text{(Eq 1)}$$

where Δ = vertical deflection at tip, in.

F = vertical force, lb

L = Length of cantilever, in.

The expression of the end deflection of a simple cantilever beam loaded at the end is

$$\Delta = \frac{1}{3}\frac{FL^3}{EI} \qquad \text{(Eq 2)}$$

where:

E = Young's Modulus or modulus of elasticity, psi

I = moment of inertia, $in.^4$

The moment of inertia (I) may be expressed in terms of line internal radius and wall thickness

$$I = \pi r^3 t \qquad \text{(Eq 3)}$$

Assuming wall thickness to be dictated by the allowable hoop stress(s)

$$t = \frac{Pr}{S} \qquad \text{(Eq 4)}$$

Substituting Equation 4 in Equation 3 yields

$$I = \frac{\pi r^4 P}{S} \qquad \text{(Eq 5)}$$

Substituting Equation 5 in Equation 2 yields

$$\Delta = \frac{1}{3}\frac{FL^3 S}{\pi P r^4 E} \qquad \text{(Eq 6)}$$

Finally substituting Equation 6 in Equation 1 yields

$$f = \frac{\Delta}{L} = \frac{F}{3\pi P}\left(\frac{L}{r^2}\right)^2\left(\frac{S}{E}\right) \qquad \text{(Eq 7)}$$

Expressed in terms of line diameter

$$f = \frac{16F}{3\pi P}\left(\frac{L}{d^2}\right)^2\left(\frac{S}{E}\right) \qquad \text{(Eq 8)}$$

where d = 2r = mean diameter

Considering k_1 to represent a numerical constant, which is a function of the particular beam configuration, Equation 8 may be expressed

$$f = \frac{k_1 F}{p}\left(\frac{L}{d^2}\right)^2\left(\frac{S}{E}\right) \qquad \text{(Eq 9)}$$

From this simplified study, a performance factor, PF, was derived to relate the flexibility and weight properties of a material to the geometric and pressure requirements of a specific application. This performance factor is derived as follows:

$$PF = \frac{f/F}{W/L} \qquad \text{(Eq 10)}$$

where PF = performance factor

W = weight, lb

Because the weight of the line is a function of material density and the volume of the line itself:

$$W = \pi dtL\rho = 2\pi rtL \qquad \text{(Eq 11)}$$

where ρ = density, lb/in^2.

Again expressing wall thickness in terms of allowable hoop stress by substituting Equation 4 in Equation 11

$$W = \frac{2\pi r^2 PL\rho}{S} \qquad \text{(Eq 12)}$$

Substituting Equations 1 and 12 in Equation 10 yields

$$PF = \frac{64F}{6\pi^2 P^2} \left(\frac{L}{d^3}\right)^2 \left(\frac{S^2}{\rho E}\right) \qquad \text{(Eq 13)}$$

or

$$PF = \frac{k_2 F}{P^2} \left(\frac{L}{d^3}\right)^2 \left(\frac{S^2}{\rho E}\right) \qquad \text{(Eq 14)}$$

Where k_2 is a numerical constant based on the particular beam configuration.

Inspection of Equations 9 and 14 shows the dependence of flexibility (f) and performance factor (PF) on pressure, the geometric parameters (L and d), and the material properties (S, E, and ρ).

The results may be summarized by saying that flexibility is proportional to allowable hoop tensile stress over longitudinal elastic modulus

$$f \sim \frac{S}{E}$$

and flexibility per unit weight (performance factor) is proportional to the square of allowable stress over density and elastic modulus

$$PF \sim \frac{S^2}{\rho E}.$$

Further study is indicated to determine reasonable practical limits for maximum S/E ratios. Obvious disadvantages of a line that is too flexible are excessive changes in volume with pressure or the inability of the line to maintain structural integrity under loads applied externally. When only pressure is considered, the allowable stress needs to be approximately one half that of the hoop stress.

APPENDIX H

HEAT EXCHANGER DESIGN FOR AJ-1 ENGINE

I. INTRODUCTION

This appendix presents a specific application of the heat exchanger design methods, which were examined parametrically in Section IX, B. The end result of this analysis is the heat exchanger weight. However, other considerations, such as cost and reliability, must also be examined before the optimum heat exchanger system can be selected. Therefore, no attempt has been made to optimize heat exchanger weight. This example does demonstrate that no major technical problems exist in the design of pressurant heat exchangers for large, high-pressure engines. Although the parametric analysis can be utilized, the calculation procedure will be repeated for completeness.

The AJ-1 engine system is generally representative of the majority of engines considered in this program; therefore, it was selected for the specific heat exchanger design. The heat exchanger location was selected at the turbine exhaust position because of geometry limitations of the AJ-1. The heat exchanger unit consists of two sections in series; one for heating hydrogen and one for heating oxygen.

Nomenclature is tabulated at the end of this appendix.

II. DISCUSSION

A. INPUT DATA

The following input data for the analysis correspond to the AJ-1 engine characteristics (Table IX-B-1) and reflect the parameter solution criteria established for the parametric analysis discussed in Section IX, B.

Hydrogen

Mass rate of flow (w_1)	5.5	lb/sec
Pressure (P_1)	4100	psia
Allowable pressure loss (ΔP_1)	$\sim$ 400	psia
Inlet temperature (T_{in_1})	40	°R
Exit temperature (T_{ex_1})	560	°R
Bulk Properties:		
Density (ρ_1)	2.1	lb/ft^3
Specific heat at constant pressure (c_{p_1})	4.2	Btu/lb-°R
Thermal Conductivity (k_1)	0.102	Btu/hr-ft-°R
Viscosity (μ_1)	0.505×10^{-5}	lb/ft-sec
Prandtl number (Pr_1)	0.7486	

Oxygen

Mass rate of flow (w_1)	16.5	lb/sec
Pressure (P_1)	4100	psia
Allowable pressure loss (ΔP_1)	$\sim$ 400	psia
Inlet temperature (T_{in_1})	170	°R
Exit temperature (T_{ex_1})	960	°R
Bulk Properties:		
Density (ρ_1)	22.48	lb/ft^3
Specific heat at constant pressure (c_{p_1})	0.355	Btu/lb-°R
Thermal conductivity (k_1)	0.0275	Btu/hr-ft-°R
Viscosity (μ_1)	2.05×10^{-5}	lb/ft-sec
Viscosity at tube wall temperature of 700°R (μ_s)	2.10×10^{-5}	lb/ft-sec
Prandtl number (Pr_1)	0.9530	

Hot Gas

Mass rate of flow (w_g)	500	lb/sec
Pressure (P_g)	3620	psia
Allowable pressure loss (ΔP_g)	45	psia
Inlet temperature (T_{in_g})	1800	°R
Film Properties:		
Density (ρ_g)	0.7664	lb/ft^3
Specific heat at constant pressure (c_{p_g})	1.9385	Btu/lb-°R
Thermal conductivity (k_g)	0.2393	Btu/hr-ft-°R
Viscosity (μ_g)	1.837×10^{-5}	lb/ft-sec
Prandtl number (Pr_g)	0.5363	

Tube

Material	Inconel 718	
Factor of safety (FS_w); engine design of 2.0 and tube bend of 1.335	2.67	
Inside diameter (d_i)	0.50	in.
Material Properties:		
Density (ρ_w)	0.29	lb/ft^3
Thermal conductivity (k_w)	110	Btu-in./hr-ft^2-°R
Yield strength at 0.2% offset (δ_y)	164,000	$lb/in.^2$
Modulus of elasticity (E)	23.5×10^6	$lb/in.^2$
Thermal expansion coefficient (α)	8.4×10^{-6}	in./in.
Absolute roughness of surface (ϵ)	0.000005	in.

B. HEAT EXCHANGER OPTIMIZATION

The interrelationships existing between the numerous factors, including pressurant velocities, hot gas velocities, pressure losses, etc., require a series of iterative steps to establish a complete set of design data.

To minimize the number of "trial-and-error" calculations required to optimize this design, an analysis was made of all variable factors to determine those least influenced by variations in fluid velocities and tube arrangements. The pressurant and hot gas friction factors are two such factors that are not significantly affected by these two variations.

The pressurant friction factor can be represented by

$$f_i = f_s \left[Re \left(\frac{d_i}{D_c} \right)^2 \right]^{0.05} \qquad \text{(Eq 1)*}$$

and f_i is essentially constant at a value of 0.015 for the high Reynolds numbers experienced for both hydrogen and oxygen.

The hot gas friction factor (f_o) which was determined by extrapolating data presented in Kays and London** is also essentially constant, but at a value of 0.05.

Since the heat exchanger weight is only one of the factors to be considered in an optimization process, emphasis was also placed on design simplicity and ease of fabrication, which is reflected in cost savings. These considerations are best satisfied by:

(1) Equal length of tubes per bank.
(2) Integral number of tubes.

*H. Ito, "Friction Factors for Turbulent Flow in Curve Pipes," ASME Paper No. 58--SA-14, 21 February 1958 (Unclassified).

**W. M. Kays and A. L. London, Compact Heat Exchangers, The National Press, Palo Alto, Calif., 1955.

(3) Constant tube diameter and wall thickness.

(4) Same shell diameter for both hydrogen and oxygen sections.

The hot gas exit temperature from the heat exchanger can be calculated from a heat balance of the system that results in the following equations and solutions:

Hydrogen

$$Q_1 = 3.6 \times 10^3 \, w_1 \, c_{p_1} \, (\Delta T)_1 \qquad \text{(Eq 2)}$$

$$= (3.6 \times 10^3)(5.5)(4.2)(560 - 40)$$

$$= 43.24 \times 10^5 \text{ Btu/hr}$$

Oxygen

$$Q_1 = (3.6 \times 10^3)(16.5)(0.355)(960 - 170)$$

$$= 16.66 \times 10^6 \text{ Btu/hr}$$

Therefore, the total quantity of heat transferred from the hot gas is

$$Q_1 = (43.24 + 16.66) \times 10^6$$

$$= 59.9 \times 10^6 \text{ Btu/hr}$$

The hot gas was selected to pass through the hydrogen section and then the oxygen section of the heat exchanger. Thus the hot gas exit temperature for each section is:

Hydrogen

$$T_{ex_g} = T_{in_g} - \frac{Q_1}{3.6 \times 10^3 \; w_g \; c_{p_g}} \qquad \text{(Eq 3)}$$

$$= 1800 - \frac{43.24 \times 10^6}{(3.6 \times 10^3)(500)(1.9385)}$$

$$= 1787.6°R$$

Oxygen

$$T_{ex_g} = 1787.6 - \frac{16.66 \times 10^6}{(3.6 \times 10^3)(500)(1.9385)}$$

$$= 1782.8°R$$

The remaining inlet and exit temperatures are known. Therefore, the log mean temperature (used for surface area calculations) of each section can be calculated.

Hydrogen

$$\text{LMTD} = \frac{(T_{ex_g} - T_{in_1}) - (T_{in_g} - T_{ex_1})}{\ln\left[\dfrac{T_{ex_g} - T_{in_1}}{T_{in_g} - T_{ex_1}}\right]} \qquad \text{(Eq 4)}$$

$$= \frac{(1787.6 - 40) - (1800 - 560)}{\ln\left[\dfrac{1787.6 - 40}{1800 - 560}\right]}$$

$$= 1483.3°R$$

Oxygen

$$LMTD = \frac{(1782.8 - 170) - (1787.6 - 960)}{\ln \left[\frac{1782.8 - 170}{1787.6 - 960}\right]}$$

$$= 1176.7°R$$

The minimum tube wall thickness and resulting outside tube diameter are represented by

$$d_o = d_i + 2\,\tau \qquad \text{(Eq 5)}$$

where $\tau = \dfrac{FS_w \, P_1 \, d_i}{2\,\sigma_y}$

Therefore $d_o = 0.50 + 2 \left[\dfrac{(2.67)(4100)(0.50)}{(2)(164{,}000)}\right]$

$$= 0.50 + 2\ (0.0167)$$

$$= 0.533 \text{ in.}$$

Sufficient information is known to solve directly the heat balance and LMTD equations. The remaining relationships necessary to establish the weight, however, must be solved in an iterative manner. The average heat transfer coefficients of the fluids are evaluated as follows:

Hydrogen

$$h_1 = 0.023 \frac{k_1}{12\ d_i} \left[Re\right]_b^{0.8} \left[Pr_1\right]_b^{0.4} \left[\frac{T_1}{T_w}\right]_b^{0.34} \qquad \text{(Eq 6)}$$

$$= \frac{(0.023)(0.102)}{(12)(0.50)} \left[\frac{(2.1)(0.50)\ V_1}{(12)(0.505 \times 10^{-5})}\right]^{0.8} \left[0.7486\right]^{0.4} \left[\frac{300}{600}\right]^{0.34}$$

$$= 0.674\ (V_1)^{0.8}$$

Oxygen

$$h_l = 0.027 \frac{k_l}{12\ d_i} \left[Re \right]_b^{0.8} \left[Pr_l \right]_b^{1/3} \left[\frac{\mu}{\mu_s} \right]_b^{0.14} \qquad \text{(Eq 7)}$$

$$= \frac{(0.027)(0.0275)}{(12)(0.50)} \left[\frac{(22.48)(0.50)\ V_l}{(12)(2.05 \times 10^{-5})} \right]^{0.8} \left[0.9530 \right]^{1/3} \left[\frac{2.05 \times 10^{-5}}{2.10 \times 10^{-5}} \right]^{0.14}$$

$$= 0.65\ (V_l)^{0.8}$$

Hot Gas

$$h_g = 0.0348 \frac{k_g}{12\ d_o} \left[Re \right]_f^{0.805} \left[Pr \right]_f^{1/3} \qquad \text{(Eq 8)}$$

$$= \frac{(0.0348)(0.2393)}{(12)(0.533)} \left[\frac{(0.7664)(0.533)\ V_g}{(12)(1.837 \times 10^{-5})} \right]^{0.805} \left[0.5363 \right]^{1/3}$$

$$= 0.462 (V_g)^{0.805}$$

The overall heat transfer coefficient combines the fluid heat transfer resistances with the tube wall resistance. This relationship based on the inside surface of the tube can be represented by

$$U = \frac{1}{R_l + R_w + R_g} \qquad \text{(Eq 9)}$$

where $R_l = 1/h_l$

$$R_w = \frac{144\ d_i\ \ln\ (d_o/d_i)}{2\ k_w}$$

$$R_g = (d_i/d_o)\ \frac{1}{h_g}$$

Therefore

Hydrogen

$$U = \frac{1}{\left[\dfrac{1}{0.674(V_1)^{0.8}}\right] + \left[\dfrac{(144)(0.50)\ln\ (0.533/0.50)}{(2)(110)}\right] + \left[\dfrac{0.50}{(0.533)(0.462)V_g^{\,0.805}}\right]}$$

$$= \frac{1}{(1.484\ V_1^{\,-0.8}) + (21.47 \times 10^{-3}) + (2.03\ V_g^{\,-0.805})}$$

Oxygen

$$U = \frac{1}{\left[\dfrac{1}{0.65(V_1)^{0.8}}\right] + \left[\dfrac{(144)(0.50)\ln\ (0.533/0.50)}{(2)(110)}\right] + \left[\dfrac{0.50}{(0.533)(0.462)V_g^{\,0.805}}\right]}$$

$$= \frac{1}{(1.539\ V_1^{\,-0.8}) + (21.47 \times 10^{-3}) + (2.03\ V_g^{\,-0.805})}$$

The relationship of the fluid velocities and surface area requirements becomes

Hydrogen

$$A_s = \frac{Q_1}{\text{LMTD}\ U} \qquad \text{(Eq 10)}$$

$$= \frac{43.24 \times 10^6}{1483.3}\left[1.484\ V_1^{\,-0.8} + 21.47 \times 10^{-3} + 2.03\ V_g^{\,-0.805}\right]$$

$$= (4.326\ V_1^{\,-0.8} + 0.06259 + 5.918\ V_g^{\,-0.805}) \times 10^4$$

Oxygen

$$A_s = \frac{16.66 \times 10^6}{1176.7} \left[1.539\, V_l^{-0.8} + 21.47 \times 10^{-3} + 2.03\, V_g^{-0.805} \right]$$

$$= (2.179\, V_l^{-0.8} + 0.0304 + 2.874\, V_g^{-0.805}) \times 10^4$$

The surface area as a function of the pressurant and hot gas velocities is shown in curves (a) and (b) of Figure 1. The cross-hatched zones represent hot gas velocities above the maximum allowable of 1680 ft/sec (Mach 0.3).

The relationship of pressurant velocity and pressure loss is represented by

Hydrogen

$$\Delta P_l = f_i \left[\frac{L_B}{d_i} \right] \left[\frac{\rho_l \quad V_l^2}{2\, g_c \quad 144} \right] \qquad \text{(Eq 11)}$$

$$= \frac{(0.015)\ (2.1)\ \ L_B\ V_l^2}{(0.50)\ \ (2)\ (32.2)(144)}$$

$$= 6.8 \times 10^{-6}\ L_B\ V_l^2$$

Oxygen

$$\Delta P_l = \frac{(0.015)\ (22.48)\ \ L_B\ V_l^2}{(0.50)\ (2)\ (32.2)\ (144)}$$

$$= 7.3 \times 10^{-5}\ L_B\ V_l^2$$

These pressurant pressure losses are plotted in curves (c) and (d) of Figure 1 as a function of tube-bank length and pressurant velocity. The cross-hatched zones represent pressurant velocities above the maximum allowable of Mach 0.3, assuming no pressurant pressure loss.

Figure 1 also shows the relationship between the pressurant velocities and the number of tubes. This relationship is represented by

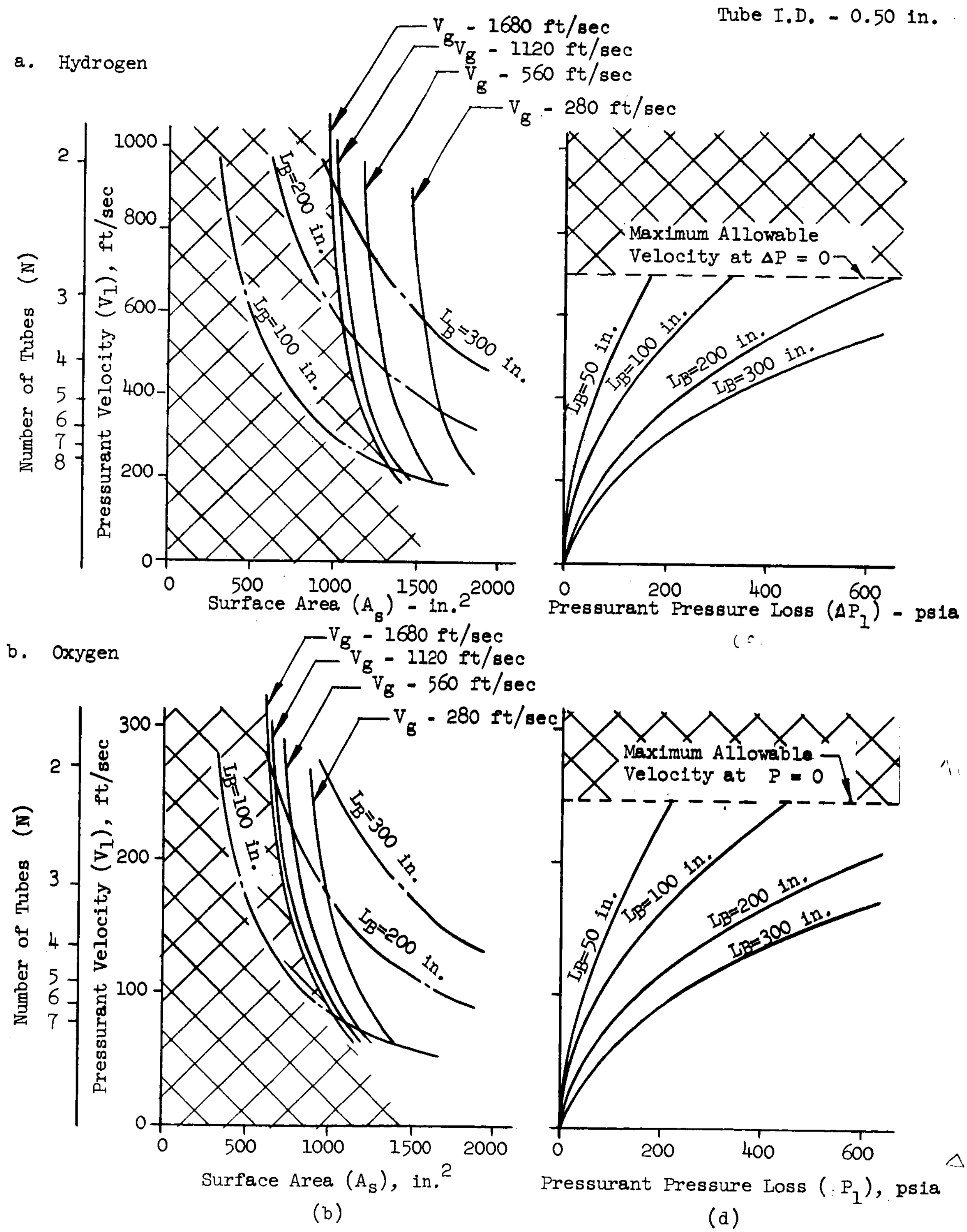


Figure 1. Pressurant Velocity vs Pressurant Pressure Loss and Surface Area for AJ-1 Engine Heat Exchanger

$$N = \frac{144\, w_1}{\rho_1 V_1 A_x} \qquad \text{(Eq 12)}$$

where $$A_x = \frac{\pi d_i^2}{4}$$

Therefore

Hydrogen

$$N = \left[\frac{(144)(5.5)(4)}{(2.1)(3.14)(\overline{0.50})^2}\right] {}^1/_{V_1}$$

$$= 1922\ ({}^1/_{V_1})$$

Oxygen

$$N = \left[\frac{(144)(16.5)(4)}{(22.48)(3.14)(\overline{0.50})^2}\right] {}^1/_{V_1}$$

$$= 538.57\ ({}^1/_{V_1})$$

Another relationship between pressurant velocities and surface area requirements is represented by the tube length per bank, i.e.,

$$A_s = \pi d_i\ L_T \qquad \text{(Eq 13)}$$

where $L_T = N\ L_B$

Therefore $A_s = \pi d_i\ N\ L_B$

$$= (3.14)(0.50)\ N\ L_B$$

$$= 1.57\ N\ L_B$$

This relationship is plotted in curves (a) and (b) Figure 1.

The hot gas pressure loss is represented by

$$\Delta P_g = f_s \left[\frac{A_s}{A_c}\right] \left[\frac{\rho_g V_g^{\,2}}{2\, g_c\, 144}\right] \qquad \text{(Eq 14)*}$$

where A_s is now the combined surface areas of the hydrogen and oxygen sections and A_c is the minimum free flow area across the bank of tubes, i.e.,

$$A_c = \frac{144\, w_g}{\rho_g V_g}$$

This pressure loss equation does not consider the shell effect on flow dynamics, because this effect is very small where a large portion of the shell's cross section is occupied by tubes.* The shape, size, spacing, and configuration of the tubes are the main factors affecting the mechanism of flow in heat exchanger shells.

Therefore

$$\Delta P_g = \frac{(0.05)(\overline{0.7664})^2 \; A_s V_g^{\,3}}{(2)(32.2)(500)(\overline{144})^2}$$

$$= 4.4 \times 10^{-11}\, A_s\, V_g^{\,3}$$

This hot gas pressure loss is plotted in Figure 2 as a function of total heat exchanger surface area and the hot gas velocity. The cross-hatched zones represent hot gas velocities above the maximum allowable of 1680 ft/sec (Mach 0.3) and pressure losses above the maximum allowable of 45 psia.

The information presented in Figures 1 and 2 represents a graphical solution to the following set of nonlinear simultaneous equations.

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* J. G. Kundsen and D. L. Katz, Fluid Dynamics and Heat Transfer, McGraw-Hill Book Company, Inc., New York, 1958.

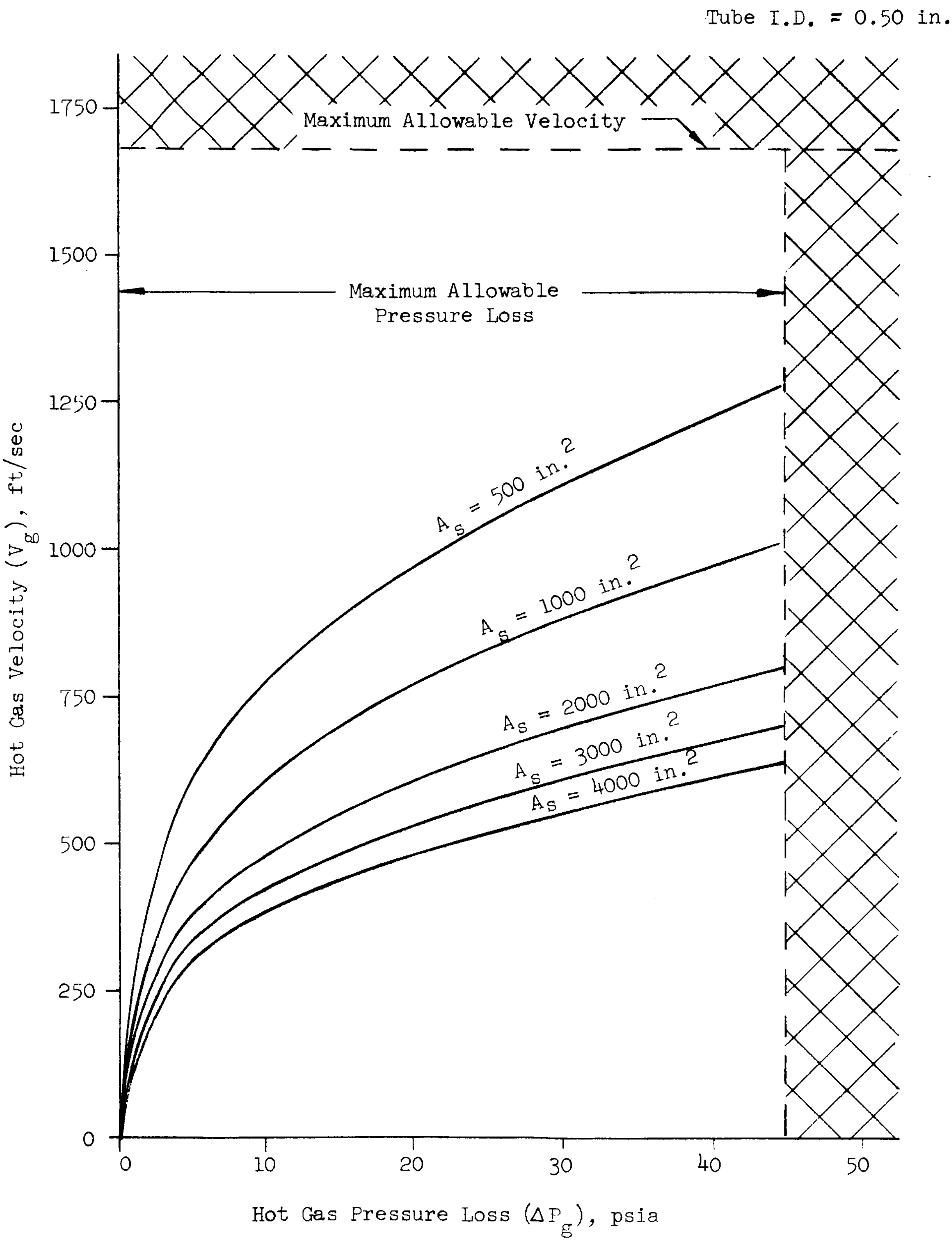


Figure 2. Hot Gas Velocity vs Hot Gas Pressure Loss for AJ-1 Engine Heat Exchanger

Hydrogen

$$A_s = (4.326\ V_1^{-0.8} + 0.06259 + 5.918\ V_g^{-0.805}) \times 10^4$$

$$N = 1922\ V_1^{-1}$$

$$A_s = 1.57\ N\ L_B$$

Oxygen

$$A_s = (2.179\ V_1^{-0.8} + 0.0304 + 2.874\ V_g^{-0.805}) \times 10^4$$

$$N = 538.57\ V_1^{-1}$$

$$A_s = 1.57\ N\ L_B$$

Both Hydrogen and Oxygen

$$\Delta P_g = 45 = 4.4 \times 10^{-11}\ A_s\ V_g^3$$

in which A_s is the combined surface areas of the hydrogen and oxygen sections. Thus

$$A_{s\ \text{both}} = A_{s\ \text{hydrogen}} + A_{s\ \text{oxygen}}$$

These eight equations contain the following variables:

Hydrogen - A_s, V_1, V_g, N and L_B

Oxygen - A_s, V_1, V_g, N and L_B

Both Hydrogen and Oxygen - A_s and V_g

The design considerations discussed earlier dictate that two of these variables (V_g and L_B) are equal for both sections. Therefore the total number of variables is reduced to nine. The number of equations can be reduced to three with four unknowns by writing each equation in terms of N, L_B, and V_g. They are:

Hydrogen

$$1.57\ N\ L_B = \left[4.326(1922\ N^{-1})^{-0.8} + 0.06259 + 5.918\ V_g^{-0.805}\right] \times 10^4$$

$$= (0.01018\ N^{0.8} + 0.06259 + 5.918\ V_g^{-0.805}) \times 10^4$$

Oxygen

$$1.57\ N\ L_B = \left[2.179(538.57\ N^{-1})^{-0.8} + 0.0304 + 2.874\ V_g^{-0.805}\right] \times 10^4$$

$$= (0.01433\ N^{0.8} + 0.0304 + 2.874\ V_g^{-0.805}) \times 10^4$$

Both Hydrogen and Oxygen

$$1.57\ L_B\ (N_{hydrogen} + N_{oxygen}) = 1.023 \times 10^{12}\ V_g^{-3}$$

These three equations with four unknowns represent an underdetermined system in which one variable may be arbitrarily selected. However, since N does not vary continuously but rather is a discrete integer, the number of possible solutions is limited. Using the information presented in Figures 1 and 2, the iteration procedure utilized to obtain a solution is as follows.

Step 1. An integral number of tubes (N) for each section was selected. Since maximum pressurant velocity and associated minimum surface area and weight were desired, the minimum number of tubes consistent with criteria established for maximum allowable velocity (see Section IX,B) was selected. This number represented three tubes for each section (Curves (c) and (d) of Figure 1).

Step 2. Then a hot gas velocity (V_g) less than Mach 0.3 (1680 ft/sec) was assumed. This assumption, combined with the previously selected number of tubes and pressurant velocity, provided surface area requirements for each section (Curves (a) and (b) of Figure 1).

Step 3. The surface area requirements were added to establish the total heat exchanger surface area. This total area was compared with the assumed hot gas velocity (Figure 2) to determine if the maximum allowable pressure loss had been exceeded. If so, a new hot gas velocity was assumed; and the "trial-and-error" procedure reverted to Step 2.

Step 4. On satisfying the hot gas pressure loss limitation, the hydrogen and oxygen tube numbers and surface area requirements were compared to determine if the tube lengths were the same for each heat exchanger section. If not, the "trial-and-error" procedure reverted to Step 1 with the selection of another combination of the tube numbers.

The completion of the iteration process results in the following design considerations:

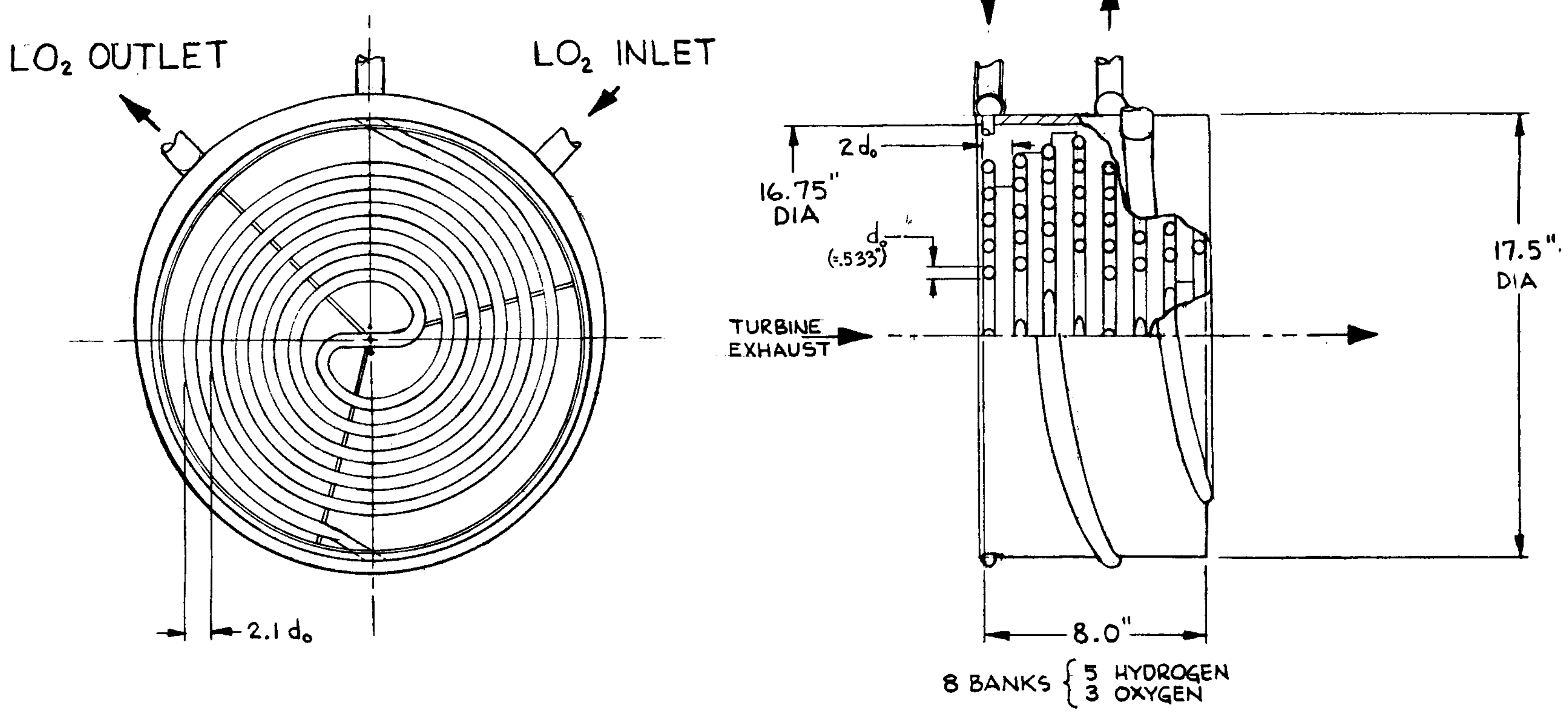
(1) Five hydrogen tubes (V_1 = 384 ft/sec).
(2) Three oxygen tubes (V_1 - 179.4 ft/sec).
(3) Hot gas velocity of 760 ft/sec.
(4) 180 in. of tubing per bank.

The resulting pressure losses were:

(1) Hydrogen pressure loss of 185 psia.
(2) Oxygen pressure loss of 420 psia.
(3) Hot pressure loss of 45 psia.

The staggered tube arrangement shown in Figure 3 was selected over the more conventional aligned arrangement, because the former provides better heat transfer with almost no additional pressure loss.* The transverse and longitudinal spacings

* E.R.G. Eckert and R.M. Drake, Jr., Heat and Mass Transfer, McGraw-Hill Book Company, Inc., New York, 1959.



(Designed to be installed as shown in Figure VI-B-22.)

Figure 3. AJ-1 Engine Heat Exchanger

of the tubes (i.e., the ratios of the transverse pitch and longitudinal pitch, respectively, to the outside diameter of tubes) were established after examining experimental data showing flow patterns for various tube spacings. These data indicated that a transverse spacing (X_t) of 2 and longitudinal spacing (X_l) of 2 were optimum. However, the transverse spacing for this design was increased to 2.1 to allow the tube bank to exit 180 degrees from the entrance, thereby providing for similarity between the "spiral-in" and "spiral-out" halves of the tube bank. This similarity reduces fabrication costs.

The shell dimensions are fixed by the tube bundle configuration and specified hot gas flow velocity. The cross sectional area of the shell is equal to the combined projected area of a tube bank and minimum free-flow area corresponding to the gas velocity. It is represented by

$$A = L_B d_o + A_c \qquad \text{(Eq 15)}$$

$$= (180)(0.533) + \frac{(144)(500)}{(0.7664)(760)}$$

$$= 220 \text{ in.}^2$$

Therefore the inside diameter of the shell is

$$D_i = \sqrt{\frac{4A}{\pi}}$$

$$= \sqrt{\frac{(4)(220)}{3.14}} \qquad \text{(Eq 16)}$$

$$= 16.75 \text{ in.}$$

and the outside diameter (assuming Inconel 718 material) is

$$D_o = D_i + 2\phi \qquad \text{(Eq 17)}$$

where

$$\phi = \frac{FS\ P_g\ D_i}{2\sigma_y}$$

Therefore $D_o = 16.75 + 2 \left[\frac{(2)(3620)(16.75)}{(2)\ \ (164,000)}\right]$

$= 16.75 + 2\ (0.37)$

$= 17.5$ in.

The shell length is

$$L = N\ d_o\ X_1 \qquad \text{(Eq 18)}$$

$= (8)(0.533)(2)$

$= 8.5$ in.

The combined weights of the tube bundles and shell are represented by

$$W = \underbrace{\left[\frac{\pi \rho_w N\ L_B}{4}\ (d_o^{\ 2} - d_i^{\ 2})\right]}_{\text{Tube bundle}} + \underbrace{\left[\frac{\pi \rho_w\ L}{4}\ (D_o^{\ 2} - D_i^{\ 2})\right]}_{\text{Shell}} \qquad \text{(Eq 19)}$$

$$= \left[\frac{(3.14)(0.29)(8)(180)(\overline{0.533}^2 - \overline{0.50}^2)}{4}\right] + \left[\frac{(3.14)(0.29)(8.5)(\overline{17.5}^2 - \overline{16.75}^2)}{4}\right]$$

$$= \qquad 11.2 \qquad + \qquad 49.8$$

$= 61$ lb

Therefore the tube bundle and shell weights will total approximately 500 lb for all eight heat exchangers required for the AJ-1 engine. This total weight does not include support structure, header manifolds, flanges, and other essential attachments.

Of these items, the supports for mounting the tube bundles to their shells are most significant. Their size is affected by the tube bundle weight and fluid-dynamic drag created by the hot gas flowing over the tubes. Heavier tube bundles require heavier supports, thus enlarging any existing weight differential. The drag force is proportional to the gas density and square of the average hot gas velocity. Therefore heavier supports also are associated with higher hot gas velocities.

NOMENCLATURE

English Letter Symbols

A	Cross-sectional area of shell, in.^{2}
A_c	Minimum free flow area across bank of tubes, in.^{2}
A_s	Inside area of heat transfer surface, in.^{2}
A_x	Cross-sectional area of tube, in.^{2}
c_p	Specific heat at constant pressure, Btu/lb-°R
D	Diameter of shell, in.
D_c	Average diameter of tubular coil, in.
d	Diameter of tube, in.
E	Modulus of elasticity, lb/in.^{2}
FS	Engine design safety factor, dimensionless; FS_W for combined engine design and tube bend safety factors
f	Friction factor, dimensionless; f_s for flow inside straight drawn-tubing; f_i for flow inside coiled tubing; f_o for flow normal to a bank of tubes
G	Mass velocity, lb/hr-ft^2 of cross section; for flow inside tubes, $G_i = w_l/A_x$; for flow across tubes, $G_o = w_g/A_c$
g_c	Conversion factor in Newton's law; g_c = 32.2 (lb fluid) (ft)/(sec^2) (lb force)
h	Local coefficient of heat transfer, Btu/hr-in.^{2}-°R
K	Universal gas constant, ft-lb/lb-mole-°R
k	Thermal conductivity, Btu-in./hr-ft^2-°R for tube material; Btu/hr-ft-°R for fluid data
L	Length of shell, in.
L_B	Length of tubing per bank ($L_B = L_T/N$), in.
L_T	Total length of tubing, in.
LMTD	Logarithmic - mean temperature difference, °R
mw	Molecular weight, gm/gm-mole
N	Number of tubes, dimensionless

Nomenclature (cont.)

English Letter Symbols (cont.)

P	Pressure, psia
ΔP	Pressure loss, psia
Q	Rate of heat transfer, Btu/hr
q	Heat transfer rate per unit surface area, Btu/hr-in.^{2}
R	Local heat transfer resistance, hr-in.^{2}-°R/Btu
r	Radius of tube, in.
T	Absolute temperature, °R
ΔT	Absolute temperature differential, °R
U	Overall heat transfer coefficient, Btu/hr-in.^{2}-°R
V	Velocity of fluid, ft/sec
W	Weight, lb
w	Mass rate of flow, lb/sec
X_l	Ratio of longitudinal pitch to outside diameter of tubes, dimensionless
X_t	Ratio of transverse pitch to outside diameter of tubes, dimensionless

Greek Letter Symbols

α	Thermal expansion coefficient of material, in./in.
γ	Ratio of specific heats, dimensionless
ϵ	Absolute roughness of tube wall surface, in.
μ	Dynamic viscosity of fluid, lb/ft-sec; μ_s for oxygen at an average inside tube wall temperature
ν	Poisson's ratio of material, dimensionless
π	3.14, dimensionless
ρ	Density, lb/ft^3
σ_y	Yield strength of material at 0.2% offset, lb/in.^{2}
τ	Tube wall thickness, in.
ϕ	Shell wall thickness, in.

Nomenclature (cont.)

Dimensionless Values

M	Mach number
Pr	Prandtl number ($3600 \mu c_p / k$), a fluid property modulus
Re	Reynolds number ($\rho V d / 12 \mu$), a flow modulus

Subscripts

b	Bulk
ex	Exit
f	Film
g	Hot gas
i	Inside
in	Inlet
l	Pressurant
o	Outside
w	Tube wall

APPENDIX I

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